JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. IV. FEBRUARY, 1892. No. 1.

The Society is not responsible for the opinions expressed by members, but merely puts them on record.

THE UNITED STATES TORPEDO BOAT *CUSHING* AND ITS EXPERIMENTS INVESTIGATED.

By Chief Engineer Isherwood, U. S. N.

The *Cushing* is the first torpedo boat constructed for the United States Navy. It was built entirely—hull and machinery—in the shops of the Herreshoff Manufacturing Company at Bristol, Rhode Island, a firm of world-wide reputation for designing and constructing small vessels of excessively high speed, whether driven by steam power alone or by sails alone, and of exceptionally excellent materials and workmanship. This firm has introduced into both kinds of these vessels, of which it makes a specialty, many novelties, not only in the details of construction but in the application of the principles of naval architecture, and always with remarkable success. The *Cushing* is, consequently, an illustration of the best that can be done with the most superior materials and the finest workmanship combined by the scientific and practical knowledge derived from extensive study and experience. It was launched on the 23d of January, 1890. The vessel was intended to be used at sea for the discharge at a distant enemy of auto-mobile torpedoes, with

the ability to protect itself by means of its own guns against the attack of any vessel of about its own size and speed. The sphere of action of such a vessel is necessarily very limited by its small dimensions; by the comparatively little weight of coal carried, supposing the vessel to be employed at a high speed, as it would have to be in war service; by the excessive lightness of the scantling of its hull; by the almost complete occupancy of the vessel with machinery and fuel, leaving only enough displacement and space for the service of the torpedoes; and by the more or less roughness of the water in which it operates. As a consequence, such a vessel can only be used with safety and effect in water smooth or nearly so. For the combination of the highest practicable speed on the dimensions given, with the greatest facility for turning, all nautical qualities have necessarily and properly been sacrificed. The *Cushing*, therefore, must be considered from an engineering point of view as a mere racing and turning machine, restricted for satisfactory performance to very short distances in very smooth water, and in which all other properties have been subordinated to speed and manœuvering. The utility of such a vessel as a naval weapon, though constructed for sea service, seems limited to smooth water, to darkness, to a foe at anchor, and to a surprise—exceptional conditions all, and involving gross incapacity and absolute heedlessness on the part of the antagonist, supposed to be a large vessel, a single shot by which, and from the smallest gun, at a distance far greater than the radius of action of the discharged torpedo, would prove totally destructive. In a rough sea nothing could be accomplished by a vessel like the *Cushing*, even if its hull should have sufficient strength to withstand the force of the waves, for, owing to its small dimensions and to the vessel's lightness per square foot of horizontal projection of the hull, the violence of its movements, rolling, pitching, rising and falling, and the resistance of the waves, would be relatively so great as to reduce the speed below that of ordinary modern naval vessels in such a sea, while the course of a discharged torpedo would be erratic in the extreme. Now, the open sea, and generally a rough one, is the very place where naval battles will

probably be fought, and where torpedo boats most certainly cannot be used; the large vessels fighting these battles can be, and are, fitted with better torpedo appliances than any small vessels can be, and can much better use them. When auto-mobile torpedoes can be carried by and launched from the large vessels of a navy without interference with the gun service of the same, a provision of small, and therefore inefficient, torpedo boats to perform the same duty in an inferior manner, seems a *prima facie* absurdity. The speed of the torpedo boat in a rough sea being less than the speed of the modern battle ship or cruiser, it cannot close at will with its larger antagonist, and the loud humming of its "blowers" makes surprise impossible. A properly handled battle ship or cruiser attacked in the open sea by a swarm of torpedo boats would be in no more danger from them than a whale from a shoal of minnows, while their destruction would be certain. An auto-mobile torpedo, to have any chance of success, must be launched within a short distance of its object, and in *smooth water*, and there seems every reason to believe that a first-class gun fired at the same range would do quite as much damage and with greater certainty. Then why the torpedo? A boat of the *Cushing's* size would be as surely hit as a large vessel when within the short distance it must come to effectively discharge its torpedo.

The use of auto-mobile torpedoes to be discharged either from the shore or from vessels, like the use of fixed torpedoes, belongs to defensive not aggressive warfare, and to the same system belong also rams, monitors, gunboats and fortifications. These methods may have local value defensively under very favorable conditions, but are useless in deciding the fate of a war. They are like skirmishes, affairs of outposts, guerilla attacks, etc., of armies, absolutely without effect upon the issue of the campaign. A naval war is decided by only first-class ships collected into fleets and giving battle in the open sea. The victor then commands the sea, with all which that phrase implies. The business of a navy is not to attempt the capture of towns, which it can neither take nor hold, though possibly injure by distant bombardment, nor to attack fortifications, which if properly made

and defended will sink any fleet without much damage to themselves; but, by clearing the seas of the armed vessels of an enemy, to utterly destroy his commerce and blockade his ports, preventing him from sending armies abroad, or protecting his colonies, or receiving aid by sea. As long as a country has a victorious navy, neither the blockade of its ports, nor its invasion by sea, is possible, and there would be no use for torpedoes, torpedo boats, gunboats, rams, monitors, harbor fortifications, etc., and only a navy composed of larger and better seagoing vessels, defensively and offensively, and more numerous than those of its antagonist, can be a victorious one. The never-captured city of Sparta never had any walls, because, as no foe was ever permitted to get within sight of it, none were ever needed. Sparta perfectly understood that if she must fight, the stand should be made at the threshold and not on the hearthstone. The cheapest as well as the most effective means of protecting a country from sea attack, is an aggressive navy sufficiently powerful to deliver battle victoriously in mid-ocean, and no country rich enough to maintain such a navy should ever resort to other means for such a purpose, for none other can be successful. Unarmored cruisers, gunboats, rams, torpedo boats, monitors and the whole crowd of absurd substitutes for battle ships, would be simply blockaded and reduced to inaction in the shallow waters of the country imbecile enough to rely upon them for protection, and if any of them were caught outside of their harbors, their capture or destruction would be quickly effected. The great truths to be luminously apprehended are that a navy inferior to the navy of the enemy is no navy at all, that the money spent for it is so much money wasted, and that dependence on it is living in a fool's paradise.

Apart, however, from naval utility, the *Cushing* is a very interesting production to the engineer, for the small dimensions of the hull and the various combinations which the steam cylinders admit, enable a great number of engineering problems to be practically solved at comparatively little cost and with much rapidity.

As regards the hull, experiments should be made upon its

resistance when backing at different speeds, as well as when moving forwards, and also when turning in circles of different diameters. Likewise, with the same displacement but different trims of vessel, both by the bow and by the stern. The degree of "squatting" at different speeds should also be determined, and its effect upon the resistance. The backing experiments would give much information regarding the propelling efficiency of twin screws when placed in the bow, and regarding their value in that position for manœuvering.

The screws enable a very exhaustive series of experiments to be made in turning, thus establishing the general laws governing the action of twin screws in the following among other particulars: 1st. When used combined in propelling ahead with equal powers applied to each, the turning being done by the rudder alone. 2d. When one screw is used alone with the rudder amidship, the other screw remaining stationary. 3d. When one screw is used alone with the rudder amidship, the other screw revolving uncoupled by the reaction of the water against its blades. Then these three trials should be repeated with the rudder in use at various angles up to the maximum. In all cases the power exerted by the engines should be ascertained with the indicator, and the speed made by the vessel on its circles should be determined by an accurate speed register.

The form of the immersed solid of the hull is exceedingly well designed for easy turning, the lateral resistance at both ends being made as little as possible by the upward curvature of the vessel's bottom without skeg at either end, so that the draught of water at the ends is very small compared with the draught amidship.

The experiments should also determine the powers required for equal speed of vessel when one screw is used and when both screws are used; also, the angle of rudder required for preserving the course in the previous case, and the angle made by the vessel's keel with the course of the vessel's center of gravity.

As each engine is quadruple expansion and consists of five steam cylinders, and as the boilers are safe at pressures up certainly to 350 pounds per square inch above the atmosphere, a

great variety of most valuable experiments could be made with one engine alone—the vessel being secured stationary to the wharf—on the relative economy due to such excessive pressures compared to lower ones, down to any desired limit. Then, by successively detaching the small cylinder and the first intermediate cylinder, removing their valves and connecting rods, the relative economy of the quadruple expansion engine, the triple expansion engine and the compound engine could be ascertained for equal developments of power, the initial steam pressure being proportionally lessened in the latter case. A tank placed on the wharf would supply the feed water, which would be exactly measured in it, and represent the cost of the power.

As there is certainly no naval service for the *Cushing* to perform during peace, it could not be better employed than in making the experiments just briefly sketched, to which others quite as interesting and valuable might be added. The expense would be only the insignificant cost of the coal. The results would possess the highest engineering value, particularly to the navy, where the knowledge of the best type of engine, and the most economic boiler pressure to be used, is of more importance than elsewhere.

In the absence of direct and complete experiments, the writer will endeavor in this article to make what use he can of the results of such trials as have been made, and to draw from them whatever deductions their very limited data will permit, before doing which the exact dimensions of the hull and machinery must be given, together with such a description as may convey a correct idea of the same, as follows:

HULL.

The hull is constructed entirely of so-called "mild steel;" it is, of course, unarmored, and it is not sheathed with wood covered with copper. It has no keel, and it is propelled by twin screws. There is nothing which can be called a "skeg" at either end of the hull, the bottom of which from end to end approximates an elliptic arc of 138 feet chord and 4 feet versed sine, bent upwards. That is to say, the hull at its greatest im-

mersed transverse section, which is about at the center, is 4 feet below either end. This gives a kind of bateau-shaped bottom with easy turning qualities. The water lines are very sharp. The deck is, crosswise the vessel, an elliptical arc, and, for a length of 30 feet from the stem, is formed into a " whaleback." Both ends of the hull above the line of greatest length fall inwards one foot from the vertical at the deck. The frames are steel angles spaced 18 inches from center to center. The deck beams are also of steel angles. The deck and bottom plating varies from five sixty-fourths to one quarter of an inch in thickness, and was bent cold to form. The average length of these plates covers nine frames, and is 162 inches.

There are ten water-tight bulkheads crosswise the hull, dividing it into eleven water-tight compartments. Four of these bulkheads are between the stem and the forward boiler, and four are between the sternpost and the aft boiler. The longitudinal bulkheads, one on each side of the hull, are continuous for 57 feet on each side of the boilers and engines, and form the fronts of the coal bunkers abreast of the three middle compartments. The remaining two crosswise bulkheads form the three compartments which contain, respectively, the forward boiler, the engines and the aft boiler.

The third compartment from the bow, and under the " whaleback," is the torpedo room. It is also the forecastle and quarters for the crew, of whom it will accommodate twelve. The loading of the torpedoes and the charging of the launching tubes, as well as the manipulation of the breech mechanism, are done in this compartment.

The compartment immediately forward contains the auxiliary engine for the steam capstan.

The cabin is in the third compartment from the stern, and accommodates four officers. Abaft the cabin are the pantry and stores for the officers.

The machinists' quarters are in the compartment immediately forward of the one containing the cabin.

The conning towers are of one-quarter of an inch steel plate, and each contains a steam-steering and a hand-steering apparatus.

The rudder is suspended from above; it is counterbalanced and made of bronze. Its area is 13.25 square feet.

From the aft end of the hull to the center of the screws is 8 feet, at which point their axes are 4 feet apart. The hull opposite the screws is indented 5 inches. At the forward end of the engines the axes of the main shafts are 6 feet apart. When the vessel is on even keel, the axes of the shafts are horizontal.

In smooth water, twenty ventilators through the deck supply the different compartments with air. In rough water, air circulation through the compartments is obtained by connections between them and the boiler rooms and the jackets or casings of the chimnies.

All the work is done by steam power, such as hoisting anchors, boat, etc. The vessel is lighted by electricity, and the pumping out or freeing the compartments of water is done by independent steam ejectors capable of discharging 14 tons of water per minute.

The armament, exclusive of torpedoes, consists of three rapid-firing six-pounder Hotchkiss breech-loading rifles, and two or more Gatlings.

The following are the principal dimensions and proportions of the vessel, the calculations being all made for a 4.875 feet depth of immersion of the hull amidship below the water line, corresponding to a displacement of 105.3 tons, which is sensibly the displacement at which all of Mr. Nathaniel G. Herreshoff's experiments, hereinafter described, were made. At this displacement, the length of the vessel on the water line was the same as the extreme length of the vessel. During Mr. Herreshoft's experiments, the hull trimmed 2 feet 9 inches by the stern:

Extreme length of the hull, being also the length on the water line	138 feet.
Extreme breadth of hull	15.05 feet.
Extreme breadth of hull on the water line	14.25 feet.
Depth of hold	9.21 feet.
Greatest depth of hull amidship below water line	4.875 feet.
Depth of hull below water line at 2.5 feet abaft the forward end of the hull	2 feet.
Depth of hull below water line at 2.5 feet forward of the aft end of the hull	4.75 feet.
Vessel's trim by the stern	2 75 feet.
Area of greatest immersed transverse section	47.68 square feet.

Area of water section	1,300 square feet.
Immersed external or wetted surface of hull	1,805 square feet.
Wetted surface of rudder	26.5 square feet.
Total wetted surface of vessel	1,831.5 square feet.
Displacement	105.3 tons.
Displacement at water line per inch of draught	3 1013 tons.
Ratio of the extreme length to the extreme breadth on the water line	9.68421
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.68635
Ratio of the water section to its circumscribing parallelogram	0.66107
Ratio of the displacement to its circumscribing parallelopipedon	0.38369
Ratio of the displacement to a prism having the greatest immersed transverse section of the hull for base and the length of the hull for height	0.68635
Depth of hull amidship, from top of side to bottom of hull	7⅞ feet.
From top of side of hull amidship to water line	2.79 feet.

ENGINES.*

There are two duplicate motive quadruple-expansion engines directly connected to twin screws, one engine to one screw, and each engine operating independently of the other. These engines do no other work than the direct propulsion of the vessel; the pumping and air-blowing being done by separate simple steam engines.

Each engine consists of five cylinders, namely, one small or high-pressure cylinder, one first intermediate cylinder, one second intermediate cylinder, and two duplicate large or low-pressure cylinders, the large cylinder capacity being divided into two duplicate cylinders. All the cylinders are without steam jackets, and all have piston valves, not packed but working with a metallic fit. There is one valve to each cylinder. There are no independent cut-off valves, the cutting off of the steam being effected in each cylinder by steam lap on the valve, and was intended to be at 0.7 of the stroke of the piston from the commencement. The cylinders of each engine are immediately opposite the corresponding cylinders of the other engine. Each engine is vertical and direct-acting, with the axes of its cylinders in a vertical plane parallel to the central longitudinal vertical plane of the hull. The valve chests of all the cylinders project

*For engraving of engines, see JOURNAL of May, 1890, page 220

from the sides of the latter in such manner that the valves are worked directly by cranks on a special shaft having its axis parallel with the axis of the crank shaft, and receiving its motion from the latter by means of two toothed spur wheels of equal diameter geared together, one upon the crankshaft, the other upon the special shaft, which, of course, makes revolution for revolution with the crankshaft. The reversing of the engine is done by partially rotating the special shaft by means of spiral feathers and sleeves worked by hand levers. The engine works exactly the same and equally well going forward or backing. The reversing is effected almost instantaneously.

The cylinders are supported on twelve forged steel stanchions, $1\frac{1}{2}$ inches in diameter at just above the bed plate. These stanchions bolt through pillow blocks, bed plate, and strengthening plate, and are secured by lock nuts. The bed plate is of rolled steel $\frac{7}{8}$ inch thick, and supports pillow blocks, stanchions, crankshaft, and special shaft working the valves. Diagonal rods are bolted to bed plate and main stanchions. The crosshead guides are bolted to horizontal bars secured to the main stanchions.

There is one piston rod to each cylinder, and it extends through the lower head only.

There is one surface condenser, and it is in common for both motive engines, and for all the auxiliary engines; it is placed between the large cylinders of the two motive engines, and is bolted to the keelson and floor plates of the hull. It has a copper cylindrical shell tinned on the inside and outside, also tube plates of brass tinned on the inside. The exhaust steam from all engines is received into the space surrounding the tubes, which are of brass, while the injection or refrigerating water passes once directly through the tubes. The cylindrical condenser connects by conical chambers at each end with pipes leading outboard for supplying and discharging the injection water. This water is supplied to the forward chamber partly by a scoop projecting outboard, and partly by a horizontal Archimedian screw of $13\frac{3}{4}$ inches diameter, placed in and filling the pipe. The scoop is made effective by the speed of the vessel,

and the screw is worked by an independent steam cylinder of 4 inches diameter and 4 inches stroke of piston.

Each engine has three air pumps. They are vertical and single acting with bucket pistons, are each 10 inches diameter and 5 inches stroke of piston, they have no foot valves, the bucket and delivery valves are annular and of $\frac{1}{4}$-inch thick brass. The lower ends of the pumps are $2\frac{1}{2}$ feet above the bottom of the condenser. The three pumps are driven by a pair of simple engines with cylinders $3\frac{1}{2}$ inches diameter and 5 inches stroke of piston, geared to make $3\frac{1}{3}$ times as many double strokes of piston as the pumps. These engines also work the feed pumps, three in number, single acting, $2\frac{7}{8}$ inches in diameter with 5 inches stroke of plunger.

There are two blowers supplying air to the two fire rooms, which are closed air tight: one blower to each fire room and independently of the other. They are secured to the forward and aft bulkheads of the engine room respectively, are 3 feet in outside diameter of case and can be driven to 2,000 revolutions per minute. They can easily furnish with clean fires an air pressure in the fire rooms equivalent to that of a column of water 5 inches high. Each blower is worked directly by a single steam cylinder 4 inches in diameter with 4 inches stroke of piston.

When the motive engines are in operation, the exhaust steam from all the auxiliary engines is delivered into the receivers between the second intermediate cylinder and the low-pressure cylinders. When the motive engines are not in operation the auxiliary ones exhaust direct into the condenser.

The following are the principal dimensions and proportions of one engine:

Number of high-pressure cylinders	1.
Diameter of the high-pressure cylinder	$11\frac{1}{4}$ inches.
Diameter of the piston rod of the high-pressure cylinder	$1\frac{3}{4}$ inches.
Net area of the piston of the high pressure cylinder	98.1995 square inches.
Stroke of the piston of the high-pressure cylinder	15 inches.
Space displacement by the piston of the high-pressure cylinder per stroke	0.852426 cubic foot.
Space in the steam passage and clearance at one end of the high-pressure cylinder	0.112008 cubic foot

Fraction which the space in the steam passage and clearance at one end of the high-pressure cylinder, is of the space displacement by the piston of the high-pressure cylinder per stroke 0.131400
Number of 1st intermediate cylinders ..... 1.
Diameter of the 1st intermediate cylinder ..... 16 inches.
Diameter of the piston rod of the 1st intermediate cylinder ..... $1\frac{3}{4}$ inches.
Net area of the piston of the 1st intermediate cylinder ..... 199.8598 square inches.
Stroke of the piston of the 1st intermediate cylinder ..... 15 inches.
Space displacement by the piston of the 1st intermediate cylinder per stroke ..... 1.734894 cubic feet.
Space in the steam passage and clearance at one end of the 1st intermediate cylinder ..... 0.205420 cubic foot.
Fraction which the space in the steam passage and clearance at one end of the 1st intermediate cylinder, is of the space displacement by the piston of the 1st intermediate cylinder per stroke 0.118405
Number of 2d intermediate cylinders ..... 1.
Diameter of the second intermediate cylinder ..... $22\frac{1}{2}$ inches.
Diameter of the piston rod of the 2d intermediate cylinder ..... 2 inches.
Net area of the piston of the 2d intermediate cylinder ..... 396.03795 square inches.
Stroke of the piston of the 2d intermediate cylinder ..... 15 inches.
Space displacement by the piston of the 2d intermediate cylinder per stroke ..... 3.437829 cubic feet.
Space in the steam passage and clearance at one end of the 2d intermediate cylinder ..... 0.363431 cubic foot.
Fraction which the space in the steam passage and clearance at one end of the 2d intermediate cylinder, is of the space displacement by the piston of the 2d intermediate cylinder, per stroke 0.105715
Number of low-pressure or large cylinders ..... 2.
Diameter of each low-pressure cylinder ..... $22\frac{1}{2}$ inches.
Diameter of piston rod of each low-pressure cylinder ..... 2 inches.
Aggregate net area of the pistons of the two low-pressure cylinders ..... 792.0759 square inches.
Stroke of the pistons of the two low-pressure cylinders ..... 15 inches.
Aggregate space displacement by the pistons of the two low-pressure cylinders per stroke ..... 6.875659 cubic feet.
Aggregate space in the steam passage and clearance at one end of the two low pressure cylinders ..... 0.726863 cubic feet.
Fraction which the aggregate space in the steam passage and clearance at one end of the two low-pressure cylinders, is of the aggregate space displacement by the pistons of the two low-pressure cylinders per stroke ..... 0.105715 cubic feet.
Thickness of metal of all steam cylinders ..... $\frac{9}{16}$ inch.
Clearance for all steam pistons ..... $\frac{1}{4}$ inch.
Area of the steam port of the high-pressure cylinder ..... 19.6 square inches.
Area of the steam port of the 1st intermediate cylinder ..... 33 square inches.
Area of the steam port of the 2d intermediate cylinder ..... 77 square inches.

Area of the steam port of each of the two low-pressure cylinders.. 77 square inches.
Ratio of the net area of the piston of the 1st intermediate cylinder to the net area of the piston of the high-pressure cylinder..... 2.035243
Ratio of the net area of the piston of the 2d intermediate cylinder to the net area of the piston of the high-pressure cylinder..... 4.032995
Ratio of the aggregate net area of the pistons of the two low-pressure cylinders to the net area of the piston of the high-pressure cylinder..... 8.065989
Ratio of the sum of the space displacement of the piston of the 1st intermediate cylinder per stroke, and of the space in the steam passage and clearance at one end of that cylinder, to the sum of the space displacement of the piston of the high-pressure cylinder per stroke, and of the space in the steam passage and clearance at one end of that cylinder..... 2.011868
Ratio of the sum of the space displacement of the piston of the 2d intermediate cylinder per stroke, and of the space in the steam passage and clearance at one end of that cylinder, to the sum of the space displacement of the piston of the high-pressure cylinder per stroke, and of the space in the steam passage and clearance at one end of that cylinder..... 3.941441
Ratio of the sum of the aggregate space displacement of the pistons of the two low-pressure cylinders per stroke, and of the aggregate space in the steam passages and clearances at one end of those cylinders, to the sum of the space displacement of the piston of the high pressure cylinder per stroke, and of the space in the steam passage and clearance at one end of that cylinder..... 7.882885
Number of tubes in the surface condenser for both engines..... 1,022.
External diameter of the tubes in the surface condenser..... $\frac{5}{8}$ inch.
Length of the tubes in the surface condenser..... $6\frac{1}{3}$ feet.
Steam condensing surface in the tubes of the surface condenser... 1,059 square feet.
Number of crank pins..... 5.
Diameter of the first two crank-pin journals..... $4\frac{1}{2}$ inches.
Diameter of the last three crank-pin journals..... $5\frac{1}{2}$ inches.
Length of all crank-pin journals..... 8 inches.
Diameter of the first or forward crank-shaft journal..... $4\frac{1}{2}$ inches.
Length of the first or forward crank-shaft journal..... 7 inches.
Diameter of the second and third crank shaft journals..... $4\frac{1}{2}$ inches.
Length of the second and third crank-shaft journals..... 10 inches.
Diameter of the fourth, fifth and sixth crank-shaft journals..... $5\frac{1}{2}$ inches.
Length of the fourth, fifth and sixth crank-shaft journals..... 12 inches.
Diameter of the line shafting..... $5\frac{1}{4}$ inches.
Number of collars in thrust bearing..... 8.
Inside diameter of collars..... $5\frac{1}{4}$ inches.
Outside diameter of collars..... $6\frac{1}{2}$ inches.
Length of main crosshead guide gib..... $8\frac{1}{2}$ inches.
Breadth of main crosshead guide gib..... $7\frac{1}{4}$ inches.
Length of connecting rod between centers..... 37 inches.
The connecting rod is composed of two flat pieces of metal, each piece being at top $2\frac{1}{2}$ by $\frac{5}{8}$ inch, and at bottom $2\frac{5}{8}$ by $\frac{5}{8}$ inch.

Length of steam pipe to one engine	11 feet 6 inches.
Length of steam pipe to the other engine	30 feet 6 inches.
Diameter of steam pipes	5 inches.
Length of exhaust pipe	5 feet 6 inches.
Diameter of exhaust pipe on low-pressure cylinder	11½ inches.
Diameter of exhaust pipe on surface condenser	13½ inches.

BOILERS.*

The vessel has two duplicate Thornycroft boilers, one in the compartment immediately forward of the compartment containing the engines, the other in the compartment immediately abaft of the one containing the engines. These boilers are designed for forced or artificial draught, the fire rooms being made air tight, and the air pressure produced in them by two fan blowers, one to each fire room. The following are the principal dimensions of *one* boiler:

Extreme length of boiler	8 feet 4½ inches.
Extreme breadth of boiler	8 feet 4 inches.
Height of boiler at ends	8 feet 10½ inches.
Height of boiler at center	9 feet 6 inches.
Breadth of grate	5 feet ½ inch.
Length of grate	7 feet 8 inches.
Area of grate surface	38.652778 square feet.
Number of tubes (iron)	950.
Outside diameter of tubes of outside row and of adjacent row	1¾ inches.
Outside diameter of all other tubes	1⅝ inches.
Aggregate heating surface of the boiler measured on the outside of the tubes, and for half the circumference of the tubes of the outer row	2,252 square feet.
Aggregate heating surface of the boiler measured on the inside of the tubes, and for half the circumference of the tubes of the outer row	1,883 square feet.
Ratio of the heating to the grate surface, calculated for the inside circumference of tubes	48.7158
Height of chimney above the level of the grate	16 feet 9 inches.
Inside diameter of the two lower or water drums	1 foot 2 inches.
Inside diameter of the upper or steam drum	2 feet 2 inches.
Extreme length of the upper or steam drum	11 feet 6 inches.
Weight of boiler alone	8.50 tons.
Weight of water in boiler	1.75 tons.
Weight of boiler and its contained water	10.25 tons.

*For engraving of boiler see JOURNAL of May, 1890, page 176.

SCREWS.

The vessel is fitted with twin screws of manganese bronze, galvanized, and each is operated by a separate engine independently of the other. The lowest point of the disc of each screw is 2 inches below the lowest point of the hull.

The screws have an uniform pitch for every portion of their propelling surface. The front and back edges of their blades are parallel when viewed in projection on a plane parallel to the axis of the screw, except the outer 6 inches of the radius. The back edge is continued in a radial line from the hub to the circumference, but the outer angle of the front edge is rounded tangentially on a radius of 0.49 foot. The upper half of both screws turn from the vessel when the latter is going ahead.

Both screws were moulded from the same cast-iron pattern, and their pitches—measured on the castings—were sensibly the same:

The following are the principal dimensions of each screw:

Diameter of the screw	4 feet 3 inches.
Diameter of the hub	9 inches.
Pitch of the screw	8.417 feet.
Number of blades	4.
Length of the screw in the direction of its axis	0.98 foot.
Helicoidal area of the blades	9.301668 square feet.
Projected area of the blades on a plane at right angles to axis	6.11838 square feet.
Projected area of the blades on a plane parallel to axis	3.33550 square feet.
Fraction used of the pitch, in function of surface alone	0.44514
Fraction used of the pitch, in function of propelling efficiency of the surface,	0.43456

WEIGHTS OF THE *CUSHING* AT HER DEEP-LOAD DRAUGHT OF WATER.

HULL.

	Pounds.
Forward conning tower and fittings	1,800
Aft conning tower and fittings	1,400
Rudder and fittings	2,000
Hull and fittings	70,200
Total	75,400

MACHINERY.

	Pounds.
Motive engines complete on their bed plates	22,400
Surface condenser and its connections	3,700
Pumps and their connections	7,020

Line shafting and connections for the screws	10,180
Two screws and their connections	2,600
Forward boiler and its connections	26,440
Chimney of forward boiler	1,100
After boiler and its connections	26,440
Chimney of after boiler	1,100
Pumps and their connections	9,000
Water in tanks for boilers	2,240
Water in boilers, condenser, pipes, etc	8,960
Miscellaneous, tools, duplicates, etc	820
Total	122,000

CREW AND OUTFIT.

	Pounds.
Personnel (24) and their belongings	6,000
Life preservers	160
Potable water	500
Provisions	2,500
Electric revolution counters and appliances	170
Two anchors	370
Two wire cables	360
One manilla hemp cable	220
Boat and outfit	260
Other articles	460
Total	11,000

ARMAMENT.

	Pounds.
Two tubes and two torpedoes under whale back, compartment No. 2	3,125
Air reservoirs on floor of compartment No. 3	213
Two spare torpedoes under floor of compartment No. 3	932
Two air compressors on floor of compartment No. 4	503
Two torpedoes and guns on deck between chimneys	2,693
Guns on deck between forward conning tower and forward chimney	1,100
Ammunition in compartment No. 8, in magazine on floor	1,120
Miscellaneous	314
Total	10,000

COAL	77,600

RECAPITULATION.

	Pounds.
Hull	75,400
Machinery	122,000
Coal	77,600
Crew and outfit	11,000
Armament	10,000
Total	296,000

With the above weights on board, the *Cushing* would be ready to leave port for prolonged service, and would have a draught of water of 5.600 feet amidship. The displacement would be 132.300 tons; the greatest immersed transverse section, 57.96 square feet; the load-water section, 1,300 square feet; the displacement per inch of draught, 3.1013 tons; and the immersed external or wetted surface, including rudder surface, 2,035 square feet.

EXPERIMENTS MADE OVER MEASURED BASES IN NARRAGANSETT BAY BY NATHANIEL G. HERRESHOFF, TO ASCERTAIN THE SLIPS OF THE SCREWS OF THE *CUSHING* AT VARIOUS SPEEDS BETWEEN 9.5 AND 23 GEOGRAPHICAL MILES PER HOUR, WITH THE VESSEL DISPLACING 105.3 TONS.

A large number of trials of the *Cushing* were made by Nathaniel G. Herreshoff, the designer of the machinery, over two deep water measured bases in Narragansett Bay, with the view of ascertaining the slips of the screws at different speeds of vessel varying from 9.5 to 23 geographical miles per hour, both included. The displacement of the hull did not, during these trials, vary sensibly from 105.3 tons with a trim of 33 inches at the stern. The vessel had just been completed, and the bottom was perfectly clean.

The observed data are given in the following table No. 1, and they were very carefully taken. They consisted of the speed of the vessel, the number of revolutions made per minute by each of the twin screws, the steam pressure in the steam pipe adjacent to the small cylinder, and the vacuum in the condenser. The throttle valve was carried wide open in all cases, and the steam was cut off in the cylinders at the same point of the stroke of their pistons. The water was smooth and the breeze light in all of the trials. The number of revolutions made by the starboard and port screws in equal time were sensibly equal.

In Table No. 1—for facility of reference—the columns containing the observed and calculated quantities are numbered, and the experiments are designated in column 1 by letters. The headings of the columns are so fully descriptive that but little explanation is needed.

Table No. 1, containing the results of the trials made by Mr. N. G. Herreshoff in 1890, of the United States torpedo boat *Cushing* over two deep-water bases in Narragansett Bay, Rhode Island, to ascertain for different speeds of that vessel per hour the corresponding revolutions of its screws per minute, slips of those screws, steam pressure and vacuum in the condenser. During these trials the throttle valve remained wide open and the point of cutting off the steam unchanged. The vessel's displacement was sensibly 105.3 tons, and the hull trimmed 2.75 feet by the stern. Smooth water and light breezes.

The relative resistances of the vessel at the experimental speeds are determined, in the absence of indicator diagrams, by the relative squares of the slips of the screws in geographical miles per hour.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Letter designating trial.	Date of trial.	Length of the base in geographical miles of 6,086 feet.	Number of runs over the base.	Number of revolutions made by the screws per minute.	Speed of the screws in geographical miles per hour.	Speed of the vessel in geographical miles per hour.	Slip of the screws in geographical miles per hour.	Slip of the screws in per centum of their speed.	Steam pressure in steam pipe adjacent small cylinder, in pounds per square inch above atmosphere.	Vacuum in the condenser, in inches of mercury.	Squares of the speeds of the vessel (column 7) relatively, showing what would be its relative resistances according to that law at the experimental speeds.	Squares of the slips at the screws in geographical miles per hour (column 8), relatively, showing what were the relative resistances of the vessel at its experimental speeds.	Ratios of the actual resistances of the vessel at its experimental speeds to what would have been those resistances according to the law of the squares of the speeds, (column 13 divided by column 12.)
A	January 30.	3	2	145.25	12.059934	9.492864	2.567070	21.240720	30.125	23.125	1.00000	1.00000	1.00000
B	January 29.	3	2	174.00	14.438626	11.286879	3.151747	21.870135	39.00	23.880	1.41535	1.52155	1.07503
C	February 6.	1	2	212.076880	17.598069	13.957734	3.640335	20.686692	59.00	25.125	2.16192	2.02233	0.93543
D	January 30.	3	2	221 00	18.338715	14.396985	3.941730	21.494036	66.25	24.500	2.30013	2.37005	1.03040
E	February 6.	1	2	235.925690	19.577858	14.480526	5.097332	26.039042	79.25	25.185	2.36690	3.96384	1.70359
F	January 30.	3	2	255.50	21.201546	15.918164	5.283382	24.919726	90.25	24.750	2.81191	4.29335	1.51475
G	January 30.	3	2	286.25	23.753000	17.398528	6.354672	26.752909	121.25	24.625	3.35974	6.16185	1.83428
H	February 1.	3	2	313.50	26.014432	18.890243	7.124179	27.385498	149.00	26.000	3.95996	7.74443	1.95568
I	February 1.	3	2	337.00	27.964466	20.306812	7.657654	27.383516	177.00	25.500	4.57669	8.94794	1.95538
J	February 6.	1	2	351.601	29.176666	21.003018	8.153048	27.944302	200.00	24.875	4.99452	10.14998	2.0685.9
K	February 4.	1	4	358.50	29.750665	21.009622	8.221043	28.105491	196.83	24.420	4.99769	10.31285	2.10136
L	March 12.	1	4	356.987527	29.623343	21.383951	8.239702	27.815481	220.75	23.580	5.07415	10.36048	2.04173
M	March 13.	1	2	369.661932	30.674773	22.159091	8.515682	27.761190	237.37	24.500	5.44905	11.06548	2.03072
N	February 13.	1	6	362.697000	30.089350	22.235448	7.853902	26.101933	212.25	25.270	5.48663	9.41245	1.71552
O	March 24.	1	6	369.748623	30.681409	22.376309	8.305160	27.068978	235.00	24.000	5.55640	10.52509	1.89422
P	March 24.	1	6	369.691997	30.677268	22.408186	8.269082	26.955080	234.00	23.210	5.57563	10.43388	1.87133
Q	February 14.	1	4	368.241100	30.556871	22.434427	8.122444	26.581400	223.89	24.170	5.58532	10.45968	1.87271
R	March 24.	1	6	372.018010	30.870082	22.438104	8.432178	27.314872	231.00	24.000	5.58731	10.84951	1.94181
S	February 17.	1	3	378.245900	31.387076	23.202805	8.184871	26.077201	240.50	23.880	5.97409	10.23099	1.71244

The slips of the screws are given in column 8 in geographical miles per hour, that is to say, these slips are the differences between the axial speed of the form of the screw (pitch multiplied by revolutions) in geographical miles per hour (column 6), and the corresponding speeds of the vessel (column 7). Their use is to obtain the *relative* resistances of the vessel at the experimental speeds (column 7), which is done by reducing them to proportionals and squaring the latter. These squares will be found in column 13, and the correctness of the assumption that they truly express the relative resistances of the vessel will be shown hereinafter.

In column 12 will be found the squares of the speeds of the vessel (column 7), expressed proportionally, and these squares represent what would have been the resistances of the vessel had they followed that theoretical law. Consequently, the division of the quantities in column 13 by those in column 12 shows, relatively, how much the experimental resistances exceeded the theoretical ones. The quotients of these divisions are given in column 14. For instance, in experiment *F*, the division of the quantity in column 13 by that in column 12 shows the experimental resistance to have been 1.51475 times greater than the theoretical resistance given by the law of the squares.

The experimental slips of the screws in per centum of their axial speed are given in column 9. These are the quantities usually understood as the slip of the screw. They are the per centum which the quantities in column 8 are of the quantities in column 6.

All the quantities in Table No. 1 are experimental ones, and are given without the slightest correction. The irregularities in the calculated results given in columns 12, 13 and 14 are due to experimental errors, almost always unavoidable when the trials are made in tideways more or less strong and very variable, and with winds of sufficient force to make their disturbing influence sensible. The number of runs over the bases was also too few for each trial to give perfectly accurate determinations. Nevertheless, the experiments show in the most unmistakable manner the well-known fact that the resistances of the hull follow the

theoretical law of the squares of the speeds up to a moderate speed, and then gradually increase above that law, but they show also the quite novel facts that this increase has a maximum after which it gradually declines and finally ceases at speeds of about 18.5 geographical miles per hour, after which the resistances again follow the law of the squares of the speed remaining constantly at about double the theoretical resistances. Of course, these results are not given as general; they may be peculiar to the hull of the *Cushing*, and may not; but possibly all vessels, if pushed to speeds relatively as great for their dimensions as in the *Cushing's* case, might show the same phenomena. At the maximum speed the bow of the vessel rose 2 feet higher than when at rest, with a corresponding depression at the stern.

In the following Table No. 2, the experimental results in the preceding Table No. 1 have been generalized, corrected and completed, each experiment by means of all the others, and by the additions of the absolute resistance of the vessel in pounds, and of the horses-power developed by the engines, calculated according to a method hereinafter described. The table has been made for each speed of vessel, increasing by successive additions of one-half a geographical mile per hour from 9.5 to 23 miles per hour, both inclusive.

The first correction, and serving for a basis of all the other corrections and additions, is that of the slip of the screws in per centum of their axial speed, made as follows: On a straight line base, the various experimental speeds in column 7, Table No. 1, were laid off by scale as abscissæ, and upon ordinates erected at right angles to this base from the ends of each abscissa the corresponding slips of the screws in per centum of their speed, column 9, Table No. 1, were laid off by scale, and a graphic curve was passed among the free ends of these ordinates in a manner that left an equal number of ends on both sides of it. Then, new ordinates having been erected at new abscissæ corresponding to speed of vessel, commencing at 9.5 geographical miles per hour, and increasing by half a mile per hour up to 23 geographical miles per hour, the per centum slips of the screw (column c, Table No. 2) due to these new speeds (column a,

Table No. 2, containing the corrected slips of the screws of the United States torpedo boat *Cushing*, for speeds from 9.5 to 23.0 geographical miles per hour, both inclusive, increasing by half a mile per hour, and uninfluenced by wind or current. Also, the

Table No. 2, containing the corrected slips of the screws of the United States torpedo boat *Cushing*, for speeds from 9.5 to 23.0 geographical miles per hour, both inclusive, increasing by half a mile per hour, and uninfluenced by wind or current. Also, the calculated thrusts of these screws, the calculated indicated horses-power developed by the engines, and the distribution of the same, for those speeds of vessel. The calculations have been made from the data in Table No. 1, corrected according to a graphic curve representing the experimental slips of the screws in per centum of their axial speeds during the trials of the *Cushing*, made by Mr. N. G. Herreshoff with the vessel displacing 105 3 tons.

Assumed speeds of vessel per hour in geographical miles of 6,086 feet.	Corrected number of revolutions made by the screws per minute.	Corrected slips of the screws in per centum at their axial speeds.	Corrected slips of the screws in geographical miles per hour.	Squares of the assumed speeds of the vessel (column **n**).	Squares of the corrected slips of the screws in geographical miles per hour (column **d**).	Squares of the assumed speeds of the vessel, proportionally.	Squares of the corrected slips of the screws in geographical miles per hour, proportionally.	Quotients of the division of the squares of the corrected slips of the screws (column **h**), by the squares of the assumed speeds of the vessel (column **g**), proportionally.	Aggregate thrust of the two screws in pounds. $1365 \times \mathbf{h}$.	Distribution of the indicated horses-power developed by the engines.				Indicated horses-power developed by the engines.
										Horses-power expended in propelling the vessel alone.	Horses-power expended in the slip of the two screws.	Horses-power expended in overcoming the cohesive resistance of the water by the surface of the screws.	Horses-power absorbed by the friction of the load.	Horses-power expended in working the engines *fore* or *aft* unloaded.
a	**b**	**c**	**d**	**e**	**f**	**g**	**h**	**i**	**j**	**k**	**l**	**m**	**n**	**o**	**p**
9.5	145.285	21.20	2.55584	90.25	6.53232	1.00000	1.00000	1.000000	1365.000	39.8487	10.7234	8.3959	4.7820	34.8815	98.6415
10.0	153.932	21.20	2.70756	100.00	7.32804	1.10803	1.10803	1.000000	1512.461	46.4890	12.5072	9.7915	5.5774	40.7174	111.0825
10.5	162.578	21.20	2.85927	110.25	8.17699	1.22160	1.22160	1.000000	1667.484	53.8186	14.4790	11.3342	6.4586	48.5532	124.6410
11.0	171.225	21.20	3.01098	121.00	9.07513	1.34072	1.34072	1.000000	1830.055	61.6962	16.6469	13.0444	7.4424	57.3889	139.3786
11.5	179.871	21.20	3.16269	132.25	9.97248	1.46537	1.46537	1.000000	2000.230	70.7041	19.0219	14.8914	8.4235	67.2246	155.3340
12.0	188.518	21.33	3.37416	144.00	10.46618	1.59557	1.60196	1.003962	2185.769	80.6218	21.7291	16.9380	9.6015	77.1770	173.1370
12.5	197.380	21.33	3.53616	156.25	11.48640	1.73130	1.75835	1.015650	2400.809	92.2202	24.8551	19.2187	11.0558	87.9714	193.8713
13.0	206.247	21.33	3.69814	169.00	12.72135	1.87557	1.94765	1.040067	2658.474	106.2288	29.1462	22.1743	12.7448	101.4958	217.1658
13.5	215.115	21.33	3.86012	182.25	14.24062	2.01939	2.18594	1.082501	2976.983	123.5312	34.5382	26.6068	14.8119	118.3567	246.4568
14.0	223.635	22.30	4.04692	196.00	16.14448	2.17330	2.47148	1.137275	3373.570	145.1725	41.5359	32.8887	17.4188	138.4319	281.0817
14.5	226.644	22.60	4.18574	210.25	18.54800	2.34964	2.82342	1.201819	3875.868	172.7417	51.3072	41.8087	20.8751	164.8640	331.0817
15.0	239.654	23.60	4.69331	225.00	21.47041	2.49307	3.28664	1.318310	4486.264	206.6439	63.6942	52.3294	24.6917	196.2763	388.6955
15.5	247.078	24.40	5.00765	240.25	25.06650	2.66035	3.81118	1.432814	5229.561	249.1317	80.4140	64.2911	30.8695	245.1803	480.5883
16.0	254.503	25.15	5.37699	256.00	28.90235	2.83656	4.47142	1.576845	5830.488	286.7422	96.3469	77.2958	34.8555	276.5479	542.5383
16.5	262.981	25.80	5.73780	272.25	33.91547	3.01662	5.03886	1.670366	6698.544	348.8318	121.2918	98.0823	42.3997	336.8396	659.5353
17.0	271.458	26.40	6.09782	289.00	37.18341	3.20221	5.69222	1.777591	7769.880	406.0039	145.6318	118.0381	49.5141	397.8995	777.0074
17.5	280.381	26.67	6.43800	306.25	41.34490	3.39335	6.32928	1.865201	8639.467	464.7204	170.7512	138.0589	56.8479	459.2374	898.0088
18.0	289.303	27.02	6.79813	324.00	44.86494	3.59022	6.86815	1.911313	9375.025	518.6946	193.0159	156.7196	63.3597	514.4997	991.5645
18.5	298.141	27.20	6.91269	342.25	47.77700	3.79214	7.31394	1.928655	9983.528	567.7048	212.1095	172.6220	69.6099	573.5254	1080.7996
19.0	314.316	27.80	7.60870	361.00	50.39154	4.00000	7.71419	1.928655	10529.869	614.9559	229.7634	185.1692	75.3903	625.5128	1164.3626
19.5	322.795	27.20	7.28571	380.25	53.08138	4.21329	8.12999	1.928655	11091.976	664.8303	248.3931	202.0734	81.5110	677.4898	1254.3664
20.0	331.072	27.20	7.47253	400.00	55.83870	4.43213	8.54826	1.928655	11668.102	717.9932	267.9960	217.9390	87.9431	729.4871	1332.0694
20.5	339.348	27.20	7.65934	420.25	58.65550	4.65651	8.98680	1.928655	12258.928	772.4550	288.6696	234.7532	94.7534	782.4741	1340.8617
21.0	347.625	27.20	7.84615	441.00	61.50644	4.88642	9.42422	1.928655	12864.060	830.3556	310.4427	252.4971	101.8656	835.4613	1440.8617
21.5	355.902	27.20	8.03297	462.25	64.52863	5.12188	9.67936	1.928655	13483.901	891.0920	332.9355	270.8086	109.2516	888.4485	1548.1308
22.0	364.179	27.20	8.21978	484.00	67.50480	5.36088	10.34315	1.928655	14118.400	954.7186	356.7074	290.1205	117.0564	941.4357	1648.1340
22.5	372.456	27.20	8.40660	506.25	70.67592	5.61041	10.81865	1.928655	14767.459	1021.3042	381.5861	310.3411	125.2161	994.4230	1739.8715
23.0	380.732	27.20	8.59340	529.00	73.84050	5.86150	11.30479	1.928655	15431.037	1090.9355	407.6023	331.5818	133.7610	1047.4099	1874.6905

Table No. 2) were taken off from the curve by scale, and from them the slips in geographical miles per hour (column d, Table No. 2), and the number of revolutions made per minute by the screws (column b, Table No. 2,) having been calculated, the thrusts of the screws in pounds (column j, Table No. 2,) and the horses-power (columns k, l, m, n, o and p, Table No. 2,) corresponding to these new speeds were calculated in the manner hereinafter described.

The headings of the columns of Table No. 2 are so fully descriptive that but little explanation is required.

Columns g, h and i, Table No. 2, correspond to columns 12, 13 and 14 of Table No. 1, and show respectively the theoretical *relative* resistances of the vessel at the different assumed speeds in column *a* according to the law of the square of the speeds of the vessel, the experimental *relative* resistances according to the law of the square of the speeds of the slips of the screws in geographical miles per hour, and the ratio in which the experimental resistances exceed the theoretical. An examination of column *i*, Table No. 2, containing this ratio, shows that up to the speed of 11.5 geographical miles per hour the resistance of the vessel increased according to the law of the square of its speed; that from this speed the resistance increased gradually in a higher ratio until that ratio reached the maximum at 18.5 geographical miles per hour, above which it remained constant at 92.86 per centum more than was due to the law of the square of the speeds of the vessel. The resistance of the vessel from the 18.5 miles per hour upward was thus nearly in the ratio of the cube of its speed, and the horses-power required to propel it were nearly in the ratio of the fourth-power of the speed.

The very important fact is that the ratio of the resistance of the vessel should not continue to increase indefinitely after commencing, above the law of the square of the speed of the vessel, but that after a certain very high speed is reached, the resistance should thereafter increase in only the ratio of the law of the square of the vessel's speed, starting, of course, from the speed at which the increasing ratio ended.

This fact, should it prove to be general for all vessels, and not

peculiar to the *Cushing*, is an extremely important one in the economic development of excessively high speed.

Quite probably the hull of the *Cushing* rose bodily in the water during its maxmimum speeds, thus decreasing its displacement, greatest immersed transverse section, and wetted external surface, causing thereby a corresponding diminution of its resistance, a phenomenon made possible by the form of the hull and by its comparatively small weight per square foot of horizontal projection relatively to its great speed. As the resistance of the water to displacement by the hull increases in the ratio of the square of the speed of the latter, evidently there is a speed for every vessel, as ordinarily modeled, at which, could it be driven, it would leave the water by rising, and although such a speed may be unattainable, yet, evidently, for excessively high speeds and light hulls, as in the case of the *Cushing*, this cause may operate sufficiently to greatly reduce what would otherwise be the resistance.

The results of the *Cushing's* trials show that the resistance of our recently constructed large, sharply-modeled, and high-speed cruisers, should be ascertained experimentally at different speeds varying by a mile per hour from about 10 geographical miles per hour to the maximum, for the knowledge would be manifestly of great practical utility for future designs. With that view a complete set of progressive trials should be made, during which every quantity should be practically determined.

The chief difficulty lies in obtaining the vessel's speed *accurately* for moderate intervals of time, say two consecutive hours, during which a trial should last. All the trials should be made under the same circumstances, in the same place, at the same draught of water, by the same persons and with the same instruments, so that the results would be exact, at least relatively. The difficulty as regards the determination of the speed—the most important quantity to be obtained with absolute correctness, for it enters into the result in ratios of, and much higher than, the cubes relatively to the powers—is easily overcome by the use of the "speed register" which was employed in the somewhat similar experiments on the United States screw steamer

Nina when fitted with the Kunstadter steering screw, and the entire number of trials for any one vessel could then be made in a couple of days.

MANNER OF CALCULATING THE POWER DEVELOPED BY THE ENGINE OF A VESSEL FROM THE SLIP OF THE SCREW PROPELLING THE SAME.

There follows from the universal mechanical law of the equality in opposite directions of action and reaction, or of pressure and resistance, that the pressure exercised upon the water by the propelling surface of a screw in the direction of its axis aft, is exactly equal to the resistance of the vessel propelled by that screw in precisely the opposite direction forward. In fact, the pressure of the screw surface upon the water is the sole consequence of the resistance of the vessel, and, of course, exactly equals it.

If a screw, no matter of how large dimensions, be revolved alone in the water, and supposing it to have no resistances due to surface or to frictions, it would not exercise a particle of pressure upon the water, and would advance per revolution through the latter the distance of its pitch. But if this revolving screw be now attached to a vessel, it will then exercise upon the water a pressure exactly equal to the resistance of the vessel, and be that resistance little or great, the screw, instead of advancing through the water the distance of its pitch per revolution, will advance a less distance. The difference between its pitch and the distance it advances per revolution will be greater as the resistance is greater, and less as the resistance is less; but as long as there is any resistance there will be this difference, the technical name of which is "slip," and it is universally expressed in per centum of the pitch of the screw, or, what is the same thing, in per centum of the axial speed of the screw—that is, of the number of revolutions per unit of time multiplied by the pitch, except in the cases where the writer has expressed it in absolute units of speed. This latter expression, originated by the writer, and thus far used solely by him, is the key to the proper investigation of the true action of the screw as a submarine propeller, which cannot be understood without it.

For equal speeds of the same vessel under the same circumstances, the pressure of the propelling screw upon the water, or, technically, "the thrust" of that screw, is exactly the same, let its dimensions, form of surface, number and outline of blades be what they may, but the slips of such various screws will be very different.

The slip of the screw is not in the slightest degree affected by the direction of the water currents refilling the void made by the passage of the vessel through the water, for, because of the equal and opposite action of pressure and resistance, whatever effect is produced by such currents upon the screw, is equally produced upon the vessel, as both vessel and screw are secured to the same shaft. Also, as a corollary to the above, there never can be either no slip or negative slip; there will always be positive slip. Furthermore, the difference between the speed of the vessel and the axial speed of the form of the screw, is the true or real slip of the screw, that is to say, is the actual recession of the watery fulcrum beneath the pressure of the screw surface. Neither is the slip of the screw affected in the slightest degree by the depth of its immersion as long as it is submerged and acts on "solid" water.

Water is a typical liquid, and its mobility is so extreme that it yields to the slightest pressure. The strongest current equally with absolute quiescence will yield to any opposing pressure and proportionally to the latter. Again, so mobile is water, that, sensibly, the only resistance it opposes to pressure is the inertia of the mass immediately acted upon by the pressure. The displacement at a given velocity of a mass of water represented by one cubic foot, requires a pressure equal to the weight of a column of the water one square foot in cross section and of the height required by a body falling freely to obtain a final velocity equal to the given velocity of the water. Now, in the continuous displacement of water, as in the slip of a screw, there is an uninterrupted succession of such cubic feet, so that the quantity or mass of water displaced at the given velocity will not be the single cubic foot but a number of cubic feet proportional to the velocity of the displacement; hence, the pressure-producing col-

umn, calculated as just described, must be doubled in height in this case, because, for a given velocity of water there would be a quantity proportional to the velocity, so that the height of the pressure-producing column must be doubled.

For these reasons the resistance of water in function of velocity, is as the square of the velocity. The pressure required to give a known mass of any kind of matter a known velocity, is proportional to the velocity, and if the mass of matter increases in the ratio of the velocity, as with fluid matter, then the pressure must be increased again in the ratio of the velocity, making the resistance as the square of the velocity, for the multiplication of a number by itself produces the square of that number.

All liquids pressed by a surface move or yield to the pressure in a direction at right angles to the pressing surface, let the direction of movement of that surface be what it may. If the equivalent of this pressure in another direction is required, it must be obtained by means of the parallelogram of forces, the pressure in the right-angled direction representing the diagonal and the equivalent pressure being represented inversely by one of the sides, then the ratio of the diagonal to this side will be the ratio by which to multiply the right-angled pressure in order to obtain its equivalent pressure in the direction of the side, because these two lines represent directly the spaces over which the two pressures act in equal time or simultaneously, and, as the work done by the two pressures is equal, each pressure must be inversely proportional to the length of the line representing it.

Now, as the resistance of a screw-propelled vessel is exactly equalled by the pressure exerted upon the water by the propelling surface of the screw in the direction of its axis, supposed to be horizontal, and as the pressure of that surface in a direction at right angles to it can be calculated when the slip and dimensions of the screw are known, and as the pressure in the direction of the axis of the screw can be calculated from the pressure in the direction at right angles to the screw surface, there follows that the resistance of the vessel in the direction of its keel,

at any given speed, can be calculated from the dimensions of the screw and from its slip at that speed.

The data needed are the helicoidal surface of the screw in square feet, the diameter in feet of the helix containing the center of pressure of that surface, the pitch of the screw in feet, the slip of the screw in per centum of its axial speed, the number of revolutions made by the screw per second, and the weight of the water in pounds per cubic foot.

The helix containing the center of pressure of the helicoidal surface is a line considered as a surface of infinitesimal breadth representing the entire helicoidal surface, which would consequently have the properties of that helix.

If, upon the helicoidal surface, there be described equi-distant helices, representing equally wide "elements" or strips of that surface, the resistances in the direction of the axis of the screw of these "elements" or strips, will be, for a true screw of equal length from hub to periphery, in the ratio of the squares of the normal distances of the helices from the axis, or as the squares of the radii of the helices. The center of pressure of such a surface would be obtained as follows, supposing the surface to extend to the axis: Ascertain the area of a circle having the diameter of the screw; divide this area by 2, and the cube root of the quotient will be the diameter of the helix containing the center of pressure.

In all such screws, this diameter will be 0.7937 of the diameter of the screw. In a practical screw this fraction will be a little greater owing to the fact that the helicoidal surface ends at the hub instead of being carried to the axis, and the diameter of the hub in different screws is a different proportion of the diameter of the screw. But as such screws always have the outer angles of their blades rounded off, thus lessening the surface outside of the calculated helix, the effect of which is to lessen the above fraction, the two practical modifications of lessening simultaneously the surface at the axis and at the circumference of the screw, about neutralize each other, leaving the fraction given practically correct.

In the case of a true screw having the edges of its blades

parallel from hub to periphery when viewed in projection on a plane parallel to its axis, and having a hub, the center of pressure of its blades will be obtained as follows: Reduce the radii of the screw and of its hub to proportionals. Cube these proportionals and subtract the smaller cube from the greater. Divide the remainder by 2 and extract the cube root of the quotient. Add to this root the cube of the proportional radius of the hub, and the sum will be the radius of the center of pressure of the blades in fractions of the radius of the screw taken as unity.

Example: Radius of screw equals 10 feet or 1.0.
Radius of hub equals 4 feet or 0.4. Then

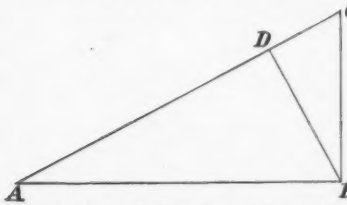
$1.0^3 =$ 1.000
$0.4^3 =$ 0.064
2)0.936
0.468 the cubic root of which

is 0.7764, adding to which the cube of 0.4 = 0.0640, there results 0.8404 for the fraction of the radius of the screw corresponding to the center of pressure of its blades.

The propelling efficiencies of such screws in function of diameter alone, all other dimensions remaining the same, will be in the ratio of the cubes of their diameters.

For screws which are not true, and have irregularly outlined or tapering blades, special methods of approximate calculation must be employed to obtain the diameter of the helix containing the center of pressure of their surface.

To show the method of applying the foregoing principles and deductions to the determination of the resistance of a screw-propelled vessel, let, in the accompanying figure, the straight line *AB* represent the circumference normal to the diameter of the helix containing the center of pressure; and let the straight line *BC*, erected at one end of *AB* at right angles to it, represent the pitch of the screw; join *A* and *C*, and from *B* let fall upon *AC* at right an-



gles to it the straight line BD. AC represents the helix containing the center of pressure, and consequently represents the helicoidal surface of the screw. BD represents the distance through which AC passes at right angles to itself while the screw makes one revolution. Suppose the slip of the screw to be 20 per centum of its axial speed, or any other fraction, then the absolute slip of AC in the direction BD would be 20 per centum of BD, or whatever fraction of BD the axial slip of the screw might be. Now, the resistance of AC in the direction BD has to be converted into its equivalent for the direction BC, which is done by multiplying it by the quotient of the division of BC by DC, DC representing the distance traveled by the pressure in that direction, while BC is the distance simultaneously traveled by the pressure in the direction BC. From the similarity of the right-angled triangles ABC and BDC, AC can be substituted for BC and BC can be substituted for DC, as $AC \div BC = BC \div DC$.

Let S represent the helicoidal surface of the screw in square feet.

Let f represent the slip in feet of the surface S in the right-angled direction to that surface during one revolution of the screw.

Let T be the number of revolutions made by the screw per second.

Let W be the weight in pounds of one cubic foot of the water acted on by the screw.

Let H be the length in feet, for one convolution of the helicoid, of the helix containing the center of pressure.

Let g be the force of gravity $= 32.175$ feet in latitude 45° at the level of the sea; and

Let P be the pitch of the screw in feet.

Then, the resistance R of the vessel in the direction of its keel, or its equivalent the thrust of the screw, will be given in pounds by the following formula:

$$R = S \times \left(\frac{f \times T}{.125\, g}\right)^2 \times W \times \frac{H}{P}.$$

The writer gives this formula for the first time, and the reason-

ings on which it is based. It is of extremely easy application, and he believes that both will be found of considerable utility in examining the results of screw propulsion, and in investigating the screw's action on water.

At first sight nothing appears so little related as the slip of a screw in per centum of its axial speed, and the resistance in pounds of the vessel propelled by it, but the connection is causal and immutable for every possible variation of condition, and the results can be calculated with certainty as long as the whole surface of the screw acts on "solid" water. All that is required to elucidate the problem, is the correct application of the indisputable principles of mechanical philosophy. Nevertheless, a word of caution is necessary here; the formula requires in an unusual degree that the data be exact, for small errors will largely affect the result.

From the resistance of the vessel or thrust of the screw, as determined by the formula, the number of indicated horses-power developed by the engine can be calculated, as follows:

1st. The number of horses-power expended in the propulsion of the hull alone is obtained by multiplying the resistance of the vessel in pounds by its speed in feet per minute, and dividing the product by 33,000.

2d. The number of horses-power expended in the slip of the screw is obtained by multiplying the number of horses-power expended in propelling the hull alone by the per centum slip of the screw, and dividing the product by the difference between 100 and that per centum.

3d. The number of horses-power expended in overcoming the cohesion of the water by the helicoidal surface on both sides of the screw blades is obtained by dividing that surface, calculated for one convolution of the helicoid, into any convenient number of equally wide "elements" or strips—the greater the number the more accurate the result—by equidistant helices, and taking the length of the central helix of each element for the length of that element. The surface of each element will be the product of the length of its central helix multiplied by its width multiplied by the fraction of the pitch used for that helix. The speed

of the element in feet per second will be obtained by multiplying the length of its central helix for one convolution by the number of revolutions made by the screw in that time. The number of horses-power required to overcome the surface resistance of the element will be given by multiplying the speed of the element in feet per second by the surface of the element in square feet (both sides of the blades) by the constant 0.45, and dividing the last product by 100 and by 550. The sum of these powers for all the elements will, of course, be the number of horses-power required to overcome the surface resistance of the screw. And, as the resistance increases or decreases in the ratio of the square of the speed of the surface, the power required to overcome it will increase or decrease in the ratio of the cube of the speed. Hence, the number of the horses-power required to overcome the surface resistance of a screw for different rotary speeds will be in the ratio of the cube of the number of revolutions made by that screw in equal time.

The constant 0.45 is the fraction of a pound experimentally required to overcome the surface resistance of one square foot of helicoid moving in its helical path with the speed of 10 feet per second. The constant divisor 100 is the square of this 10, and the 550 are the number of foot-pounds of work done per second $(\frac{33,000}{60})$ equivalent to one horse-power.

4th. There is next to be determined the number of horses-power absorbed by the friction of the load. This friction is assumed to be 7.5 per centum of the power applied to the crank pins of the engine, which is the net horses-power developed by the engines. Now the power applied to the crank pin is the sum of the powers in 1st, 2d, 3d and 4th, hence the sum of the powers in 1st, 2d and 3d (already obtained) is (100.0—7.5 =) 92.5 per centum of the net horses-power, consequently, the power in 4th will be determined by multiplying the sum of the powers in 1st, 2d and 3d by 7.5 and dividing the product by 92.5.

5th. And lastly the number of horses-power required to work the engine, *per se* or unloaded, is to be calculated. The pressure needed for this purpose in the case of the *Cushing* has been

assumed at 2.00056 pounds per square inch of the pistons of the two large cylinders; consequently the product of the multiplication of the areas of these pistons by their speed in feet per minute, and by this pressure divided by 33,000, will give the number of horses-power desired.

The indicated horses-power developed by the engine is the sum of the powers in 1st, 2d, 3d, 4th and 5th.

There remains to show numerically how the resistance of the vessel, or its equivalent the thrust of the screws in the direction of their axes was calculated from the slip of the screws in the case of the *Cushing* for the speed of vessel 9.5 geographical miles per hour (Table No. 2, top line, column j).

The center of pressure of the *Cushing's* screws has the diameter 40.8 inches, the circumference corresponding to which is 10.68 feet. Referring now to the preceding figure, the length of the line AB is 10.68 feet, the length of the line BC representing the pitch is 8.417 feet, the length of the line AC representing the helix passing through the center of pressure is 13.5981 feet, and the length of the line BD representing the distance passed through by AC in the direction at right angles to itself during one revolution of the screw is 6.6 feet. The number of revolutions made per second by the screw was $\left(\frac{145.285}{60} =\right)$ 2.42142. The slip of the screw was 21.2 per centum of its axial speed. One-eighth of 32.175 is 4.021875. The aggregate helicoidal area of the two screws was 18.603336 square feet. The weight of a cubic foot of sea water at the experimental temperature was 64 pounds. The ratio of AC to BC was $\left(\frac{13.5981}{8.417} =\right)$ 1.61555. Hence $18.603336 \times \left(\frac{6.6 \times 0.212 \times 2.42142}{4.021875}\right)^2 \times 64 \times 1.61555 = 1{,}365$ pounds, the resistance of the vessel at the speed of 9.5 geographical miles per hour.

For any other given speed of vessel under the same conditions, the resistance of the vessel in pounds will be obtained by multiplying 1,365 by the square of the quotient of the division of the speed of the slip of the screws in geographical miles per

hour when the vessel was making the given speed, by the slip in geographical miles per hour (Table No. 2, top line, column d,) when the vessel was making 9.5 geographical miles per hour. In this manner the quantities in column j of Table No. 2 were calculated.

EXPERIMENT MADE BY MR. N. G. HERRESHOFF TO ASCERTAIN THE RESULTS OF PROPELLING THE *CUSHING* BY ONE SCREW, THE OTHER REMAINING FIXED AND DRAGGING.

On the 12th of March, 1890, Mr. N. G. Herreshoff made an experiment to ascertain the results of propelling the *Cushing* with the port screw alone, the starboard screw remaining immovably fixed and dragging, while the center of gravity of the vessel was kept on the straight-line course by carrying the rudder at an angle of about 7 degrees.

The base used was in Narragansett Bay, and was 1.73513 geographical miles in length. Two runs were consecutively made over it in smooth water during a calm. The vessel had the same displacement and the same trim as during the trials in the preceding Table No. 1. The following were the results:

Steam pressure in steam pipe near the valve chest of the small cylinder in pounds per square inch above the atmosphere	186.5
Vacuum in condenser in inches of mercury	24.125
Number of revolutions made per minute by the port screw	282.602339
Speed of the vessel in geographical miles per hour	13.223062
Speed of the screw in geographical miles per hour	23.450515
Slip of the screw in geographical miles per hour	10.227453
Slip of the screw in per centum of its axial speed	43.612914

During the above performance the thrust of the single screw, calculated from its slip in the manner previously described, was:

$$\left(\frac{0.43612914 \times 6.6 \times 282.602339}{60 \times 4.021725}\right)^2 \times 64 \times 9.301668 \times 1.61555$$

= 10,924.5 pounds.

The normal resistance of the vessel at the experimental speed 13.223062 geographical miles per hour, as given in the preceding Table No. 2, was 3,048 pounds, consequently, that pressure would

be the thrust of the single screw provided its axis had been placed in the central longitudinal plane of the vessel, and had the vessel moved in the direction of that plane, and had there been no dragging starboard screw and no inclination of the rudder. But the resistance overcome by the port screw acting alone was actually ($\frac{10924.5}{3048} =$) 3.58 times as great; hence, the additional or abnormal resistances compared with the normal resistance of the vessel as unity were 2.58. This increased resistance is composed of three factors: 1st. The drag of the fixed or stationary screw. 2d. The drag of the rudder placed at the inclination of 7 degrees to the line of the keel. And 3d. The resistance due to the angle made by the keel of the vessel with the straight course of the center of gravity of the vessel. The center of gravity of the vessel moved in a straight line—the course—but the keel made a fixed angle with that line, so that the vessel was pushed forward slightly sideways, with, consequently, a correspondingly increased resistance above the normal. The bow of the vessel is deflected from the course towards the propelling screw, that is, if the port screw propels alone the bow will be on the port side of the course, and if the starboard screw propels alone the bow will be on the starboard side of the course. There is no possibility of calculating the angle thus made, and it must be obtained from observation alone, which can be easily done. There is no means of ascertaining quantitatively the separate values of the foregoing three quantities.

The above calculation of the thrust of the port screw when propelling alone is not exact, though nearly so, because the slip of the screw on which it is based is not exact. The axis of the screw and the course of the vessel should be in the same straight line to give the exact slip, whereas, in the experiment with one screw, as these lines made an angle with each other, the slip was correspondingly out of truth. The error is due to the difference in the length of the radius and secant of the angle made by the two lines, and for small angles is not material; neglecting it, the horses-power developed by the port engine, and the distribution of the same, will be as follows:

Horses-power expended in propelling the vessel alone	444.017780
Horses-power expended in the slip of the screw	343.428090
Horses-power expended in overcoming the resistance of the water to the surface of the screw blades	30.891582
Horses-power absorbed by the friction of the load	66.351685
Horses-power expended in working the engine, *per se*	33.924998
Indicated horses power developed by the engine	918.614135

The indicated horses-power required for the propulsion of the vessel normally at the above speed by two screws, are 224.5, as shown by the preceding Table No. 2, consequently $(\frac{918.614135}{224.5} =)$ 4.09 times that power was commercially needed for propelling at the same speed with one screw under the experimental conditions.

EXPERIMENT MADE BY MR. N. G. HERRESHOFF TO ASCERTAIN THE RESULTS OF BACKING THE *CUSHING* IN A STRAIGHT LINE BY BOTH SCREWS.

On the 12th of March, 1890, Mr. N. G. Herreshoff made an experiment to ascertain the results of backing the *Cushing* in a straight line over the measured base of 1.73513 geographical miles in Narragansett Bay. Two runs were consecutively made over it in smooth water during a calm. The vessel had the same displacement and the same trim as during the trials in the preceding Table No. 1. The following were the results:

Steam pressure in steam pipe near the valve chest of the small cylinder	90.50
Vacuum in condenser in inches of mercury	24.375
Number of revolutions made per minute by the screws	215.
Speed of the screw in geographical miles per hour	17 840831
Speed of the vessel in geographical miles per hour	13.900349
Slip of the screw in geographical miles per hour	3.940482
Slip of the screw in per centum of its axial speed	22.086875

The above slip of the screws is almost precisely the same that was given by them in the trials with both screws propelling the vessel forward under otherwise exactly the same conditions, namely, in going forward at the speed of 14 geographical miles per hour, the slip of the screws was 22.30 per centum of their axial speed (Table No. 2, column c), while, in backing, their

slip was 22.087 per centum. Consequently, if the screws in backing used effectively their entire surface, which supposes no reaction by them against the hull, the resistance of the latter was exactly the same be it propelled forward or backward. But if any part of the thrust of the screws reacted against the hull, thereby increasing its resistance and lessening their efficiency, then the normal resistance of the hull when going in the backward direction was, by the extent due to these causes, less than its resistance when going forward.

As there must have been some reaction of the screws against the hull, its normal resistance must have been less in the backward than in the forward direction, but as the extent of this reaction cannot be known from the data, no calculation of its effect can be made.

EXPERIMENTS MADE BY A BOARD OF U S. NAVAL ENGINEERS UNDER THE PRESIDENCY OF CHIEF ENGINEER C. H. LORING, U. S. N., TO ASCERTAIN THE ECONOMIC AND POTENTIAL VAPORIZATION OF WATER BY SEMI-BITUMINOUS COAL CONSUMED IN ONE OF THE BOILERS OF THE *CUSHING*.

Trials of the economic and potential vaporizations obtained by burning semi-bituminous coal in the furnace of one of the *Cushing's* boilers, were made by a Board of United States Naval Engineers consisting of Chief Engineers Charles H. Loring and George W. Magee, and Passed Assistant Engineers George H. Kearny and W. N. Little. The vessel was secured to the wharf of the New York Navy Yard, and the motive machinery was at rest.

The steam escape pipe, $3\frac{1}{2}$ inches in diameter, rises vertically from the main stop valve of the boiler, and has the safety valve chamber bolted in continuation upon its upper end above the deck of the vessel. This chamber was removed, and in its place was bolted to the top of the escape pipe the vertical limb of a short T pipe $3\frac{1}{2}$ inches in diameter. Then to the end of one of the horizontal arms of the T pipe was bolted the safety valve chamber, and to the end of the other horizontal arm was bolted the chamber of a screw stop valve controlled by hand. To the

chamber of this screw stop valve was bolted a copper pipe 4 inches in diameter, which, first descending vertically below the surface of the outside water, and then bending horizontally, entered an open ended wooden box or tube firmly secured in a horizontal position to the side of the vessel and having its center 20 inches below the water surface. This box was a truncated pyramid in form, 4 feet in length, 15 inches square at one end and 12 inches square at the other end. The steam was discharged from the copper pipe into the greater end of the wooden box, the combination forming a large ejector, the steam entraining a sufficient quantity of water to completely liquefy it instantaneously without producing the slightest perceptible sound.

A pot filled with sand and containing a thermometer was inserted vertically into the T pipe in order to show the temperature of the steam at that point. This temperature, however, will not be that of the steam in the boiler, but much less, depending on the diameter and length of the escape pipe. The escaping steam expands in the escape pipe throughout the entire length of the latter, does external work in displacing the atmosphere, and internal work in overcoming the cohesion of its own molecules, etc., and suffers a corresponding loss of heat which reduces its temperature.

The coal was carefully weighed on the wharf in bags filled to a uniform weight, and it was again weighed in the fire room. The greatest care was taken, by means of a system of checks, to secure the exact number of pounds consumed.

The refuse, consisting of ash, clinker and small pieces of unburnt coal falling into the ashpit from between the grate bars, was carefully collected and weighed in the dry state.

An attendant, by manipulating the screw stop valve on one end of the horizontal limb of the T pipe, kept the steam pressure in the boiler during all the trials at exactly 250 pounds per square inch above the atmosphere by the mark on the dial of the steam gauge directly communicating with the steam room. As a consequence, the temperature shown by the thermometer immersed in the steam of the T pipe remained constant during all

the trials at 398 degrees Fahrenheit, or 8 degrees less than the temperature due to the pressure of the boiler steam.

The temperature of the gases of combustion leaving the boiler was taken by two pyrometers placed in the upper portion of the chimney, with their dials above the top of the latter. After the trials were finished the pyrometers were sent to their maker, who tested them and corrected their indications.

The blower supplying the air for the combustion was bolted to the bulkhead of the fire room. Its fans were 34 inches in diameter, twelve in number, slightly curved, and each measured 10 by 10½ inches. The air was received at the center of the blower and discharged from its entire circumference, delivering into an air-tight fire room, the air pressure in which was measured by a water column in a U-shaped glass tube, one end of which communicated with the fire room, while the other was under the atmospheric pressure. The constancy of the air pressure during an experiment was maintained as nearly as possible by changes in the rotary speed of the blower.

The feed water was drawn from the mains of the Navy Yard, and was measured in its passage to the boiler by means of a Worthington meter placed on the deck of the vessel. The meter discharged its water into the open hot well of the air pumps, using that well simply as a tank, always under the atmospheric pressure. Consequently, the meter received and delivered water of uniform temperature under constant pressures, and could have no variations of rate. The rate was accurately measured. The indications of the meter were in cubic feet, from which, and the temperature of the water, the weight of the latter was calculated. The air pumps were disconnected and their communicating pipes blanked off. The auxiliary feed pumps of the engine pumped the feed water from the hot well tank into the boiler. The water supply was, of course, regulated by a screw stop valve in the feed pipe, according to signals from the fire room, and the water level in the boiler was always maintained sensibly constant throughout the experiments.

Each experiment was commenced and ended with the fire in steady action, and the same depth of coal on the grate as nearly

as could be judged by the eye. The water level in the boiler and in the hot well tank, and the steam pressure, were made precisely the same at the beginning and end of each trial. The inconveniences of the situation were so great, that no other method seemed practicable; otherwise, the weights of coal and water would have been taken from the kindling of the fire to the complete burning out of the same.

The firing was done by the firemen attached to the vessel, and in the same manner as in regular service. The equipment was thoroughly efficient, and several preliminary short trials were made to familiarize the attendants with the work and instruct them in the proper routine.

The semi-bituminous coal consumed was from the Pocahontas mine of the New River district, West Virginia. It was of excellent quality, and gave, as the mean of six analyses, when retorted, the following proximate composition by weight centesimally:

	Crude coal.	Coal exclusive of water and ash.
Coke, exclusive of ash,	75.34	77.84
Volatile matter exclusive of water,	21.09	21.79
Sulphur,	0.36	0.37
Hygrometric water,	0.92	
Ash,	2.29	
	100.00	100.00

This coal is extremely free-burning; cokes in the furnace, cakes and swells but little, and gives off a moderate quantity of dark brownish smoke.

There should be here stated that the hygrometric water in the merchantable coal is much larger that is given in the above analysis, in making which a very small sample of the coal is reduced to as fine a powder as possible, and consequently becomes very dry in the laboratory before the analysis is made. The hygrometric moisture in the coal in the dry merchantable state is about 2.5 per centum. Also, the proportion of ash in the mer-

chantable coal is much greater than in the small samples carefully selected for analysis. The true proportion of the ash as an average for large quantities of the coal, is about 4.5 per centum. The refuse drawn from the ashpits, of course, is much greater than this proportion, as it contains additionally the small pieces of unconsumed coal that may fall between the grate bars.

In the following Table No. 3 will be found the observed and calculated quantities relating to the two trials of the *Cushing's* boiler made by the Board of United States Naval Engineers. One trial, column A, Table No. 3, was made with the rate of combustion produced by an air pressure in the fire room equivalent to that of a column of water half an inch high; and the other, column B, Table No. 3, was made at the maximum rate of combustion. The blower, with the boiler pressure used, would make an average of 1,075 revolutions per minute for several hours' steaming, and produce an air pressure in the closed fire room equivalent to that of a water column 3 inches high.

With the vessel motionless at the wharf, and in a calm atmosphere, which were the conditions of the experiments, the pressure equivalent to that of a water column 3 inches high would supply air enough to the ashpits to burn 1,465 pounds of the experimental coal per hour. This was the maximum rate of combustion under these conditions, but the inference is not warranted that it would continue to be the maximum rate under other conditions. For instance, when the vessel was under way at the maximum speeds of 22 geographical miles per hour and over, there was added to the chimney draft produced by the constant fire-room pressure the draft produced by the movement of the vessel through the air, a very large addition in the case of the excessively high speeds of the *Cushing;* consequently with the vessel under way at the maximum speed and the same air pressure in the fire room equivalent to that of a water column 3 inches high, the rate of combustion would be very largely increased. This most important fact must not be lost sight of in estimating the maximum steam-producing power of the boiler.

In the case of the United States steam frigate *San Jacinto,* of which the writer was chief engineer during a three years' cruise

on the eastern coast of Asia, the boilers would burn 50 per centum more anthracite per hour during a strong monsoon ahead than during light breezes ahead.

On the last line of the following Table No. 3 is given the economic vaporization of one pound of the combustible portion of the coal in pounds of water, supposed to be vaporized under the atmospheric pressure from the temperature of 212 degrees Fahrenheit. This is not exact, owing to the boiler pressure not being the atmospheric pressure; had it been, the temperature of the steam and water in the boiler would have been 212 degrees instead of 406 degrees, and the gases of combustion would have left the boiler at a proportionately lower temperature to what they did, and the coal would have given a correspondingly increased economic vaporization of water. Taking the temperature of the gases of combustion in the furnace at 1,800 degrees Fahrenheit, the quantities on the last line of Table No. 3 should be increased by $\left(\frac{406-212}{1800}=\right)$ 0.107778, and become 11.63510 for column A and 10.66933 for column B; that is to say, these latter quantities would be the number of pounds of water that would have been vaporized per pound of the combustible portion of the coal, supposing the water to have been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the atmospheric pressure.

The above method of correction is reasonably approximate, if not precise, and is the best that can be done under the circumstances. Neither economic nor potential vaporizations can be accurately compared unless at standard temperatures of the feed water and steam. And all comparative experiments on boilers and fuel should be made under the atmospheric pressure and for the feed water temperature of 212 degrees.

This practical loss of economic and potential evaporization in the boiler due purely to increase of steam pressure, is one of the drawbacks of high steam pressure and must be always deducted from its theoretical benefits. Also, and due to the same cause, a further deduction should be made for the increased loss caused by increased radiation of heat in the case of shell boilers, for all

Table No. 3, containing the data and results of the experiments made by a Board of United States Naval Engineers on one of the boilers of the United States torpedo boat *Cushing*, to ascertain its potential and economic vaporization of water by semi-bituminous coal consumed at moderate and maximum rates of combustion by artificial draught with the vessel secured to the wharf of the New York Navy Yard.

	A Moderate rate of combustion.	B Maximum rate of combustion.
Date of the experiment	27 October, 1890.	30 October, 1890.
TOTAL QUANTITIES.		
Duration of experiment in consecutive hours	12.	12.
Total number of pounds of feed water vaporized	85,577.	132,340.
Total number of pounds of semi-bituminous coal consumed	10,526.	17,580.
Total number of pounds of refuse from the coal	846.	1,259.
Total number of pounds of combustible consumed	9,680.	16,321.
Per centum of the coal in refuse	8.0372	7.7140
RATE OF COMBUSTION.		
Pounds of coal consumed per hour	877.1667	1,465.0000
Pounds of combustible consumed per hour	806.6667	1,360.0833
Pounds of coal consumed per hour per square foot of grate	22.6935	37.9016
Pounds of combustible consumed per hour per square foot of grate	20.8696	35.1872
Fraction of a pound of coal consumed per hour per square foot of heating surface*	0.4658	0.7780
Fraction of a pound of combustible consumed per hour per square foot of heating surface*	0.4284	0.7223
PRESSURES.		
Height of barometer in inches of mercury	29.43	29.65
Atmospheric pressure in pounds per square inch above zero	14.446	14.554
Steam pressure in boiler in pounds per square inch above the atmosphere	250.000	250.000
Steam pressure in boiler in pounds per square inch above zero	264.446	264.554
Height of water column equilibrating air pressure above the atmosphere in fire room in inches	0.50	3.00
Pressure of the air in the fire room in fractions of a pound per square inch above the atmosphere	0.0180	0.1082
Number of revolutions made per minute by the blowing fan	590. (?)	1,075. (?)
TEMPERATURES.		
Temperature in degrees Fahrenheit of the external atmosphere	50.9	51.3
Temperature in degrees Fahrenheit of the air in the fire room	78.0	87.3
Temperature in degrees Fahrenheit of the feed water entering the boiler	58.5	58.0
Temperature in degrees Fahrenheit of the steam in the escape pipe, by thermometer	398.0	398.0
Temperature in degrees Fahrenheit of the steam in the boiler	406.0	406.0
Temperature in degrees Fahrenheit of the gases of combustion near top of chimney	536.5 (?)	653.0 (?)
VAPORIZATION.		
Total number of pounds of water that would have been vaporized had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the atmospheric pressure	91,052.33962	140,747.27702
Total number of pounds of water that would have been vaporized had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the atmospheric pressure	101,690.96212	157,192.29264
Pounds of water vaporized from 100 degrees Fahrenheit by one pound of coal	8.65023	8.00610
Pounds of water vaporized from 100 degrees Fahrenheit by one pound of combustible	9.40623	8.62369
Pounds of water vaporized from 212 degrees Fahrenheit by one pound of coal	9.66093	8.94154
Pounds of water vaporized from 212 degrees Fahrenheit by one pound of combustible	10.50526	9.63129

* For heating surface calculated for inner circumference of tubes.

the heat radiated from the external surfaces of such boilers has been first imparted to their contained water by the heat of the fuel and afterwards radiated from that water to objects surrounding the boiler.

THE CONTRACT TRIAL OF THE UNITED STATES TORPEDO BOAT *CUSHING*.

The builders of the *Cushing* guaranteed to the Navy Department that the vessel should make the mean speed of not less than 22 geographical miles per hour for 3 consecutive hours, consequently, on her completion, a Board composed of several line officers of the Navy, one engineer and one naval constructor, was ordered to ascertain whether this guarantee had been performed. The trial was made in Narragansett Bay, in smooth water and light breezes, and with the immersed surface of the hull perfectly clean. The vessel was run up and down the bay on a straight course an even number of times in each direction, the turnings not being counted. This course passed over the naval measured base of one geographical mile, and the vessel's speed, together with the number of revolutions made by each screw, were taken each time by chronograph, six times in all, that the vessel passed over the base. The mean displacement of the hull during the trial was 105.3 tons, and it trimmed 33 inches by the stern. The mean speed of the vessel during the three hours was determined by applying to the mean number of revolutions made per minute during that time the slip of the screws ascertained from the six runs over the one mile base. The data as regards speed of vessel and revolutions of the screws over this mile, were taken independently by Mr. N. G. Herreshoff as well as by the board. Their observations are given for experiments O and P in the preceding Table No. 1, and were almost identical.

The number of revolutions made per minute by both screws during the entire trial, and also during the passages of the vessel over the base, were so nearly the same that they may be taken to be exactly so, as their very slight difference of a very small fraction of a revolution, sometimes on the one side and sometimes on the other, may easily be due to error of observation.

* For heating surface calculated for inner circumference of tubes.

The speed of the vessel varied but little from time to time. The safety valves discharged steam at short intervals, showing the performance to be well within the power of the machinery provided. The blowers could easily have been driven to produce an air pressure in the fire room of two inches of water column greater than they did, had it been needed. In fact, the vessel could have maintained the speed of 23 geographical miles per hour, with but little more urging of the machinery.

As no indicators had been fitted to the steam cylinders, the power developed by the engines could not be measured; nor was any attempt made to ascertain the weight of coal consumed. The sole object of the trial was to determine whether the vessel could maintain for three consecutive hours the mean speed of 22 geographical miles per hour.

Throughout the entire trial there was not the slightest difficulty with any part of the machinery, which worked smoothly, noiselessly and without heating of journals. The vertical vibration of the boat was extremely small, and the transversal vibration was moderate.

About two months previous to the trial, and according to the instructions of Mr. Thornycroft, four zinc discs 14 inches in diameter and one-half an inch in thickness, were placed in each upper or steam drum of the boilers, and four round zinc bars 1⅝ inches in diameter and eight feet in length, were placed, supported, in each lower drum. After remaining in position about six weeks, the boilers being occasionally used during that time, the zinc was examined and found to be in a generally disintegrated state, about one-half of each piece retaining the original form of disc or bar, but very dark and brittle, the other half being in the form of sand or gravel and lying on the lowest part of the drums. When removed, the weight from both boilers was 380 pounds, when first placed in them the weight was 546 pounds. This zinc is kept constantly in the boilers, being renewed at short intervals, and is absolutely essential to their preservation from oxidation.

About one hour is required to raise steam from cold water when the usual quantity of wood for kindling is used. Of course,

by the use of a much greater quantity of wood, steam can be raised in about half an hour.

The coal burned on the trial was semi-bituminous from the Pocahontas mine of the New River district, West Virginia.

When the vessel left the Herreshoff wharf for the trial, the tanks containing one ton of water for supplying waste from the boilers were filled. At the end of the trial this water was entirely exhausted. The boilers were perfectly tight under the pressure, and the loss was due to the steam pumped out of the condenser by the air pump, the whole of which is always lost. The vacuum in the condenser being 24 inches of mercury, the temperature of the water of condensation, allowing for the usual proportion of air present, must have been, say, 126 degrees Fahrenheit, and the weight of steam of that temperature is 0.006 pound per cubic foot.

The following are the data and results of the trial. The speed of the vessel has been calculated from the slip 27.2 per centum of the axial speed of the form of the screw as given by the experiments in the preceding Table No. 1, generalized in Table No. 2. During the trial, the vessel "squatted" about 2 feet at the stern.

Date of the experiment	March 24th, 1890.
Duration of the experiment in consecutive hours	3.
Steam pressure in steam pipe near valve chests of small cylinders, in pounds per square inch above atmosphere	245.
Steam pressure in 1st receiver in pounds per square inch above atmosphere.	100.
Steam pressure in 2d receiver in pounds per square inch above atmosphere.	40.
Steam pressure in 3d receiver in pounds per square inch above atmosphere.	5.
Vacuum in the condenser in inches of mercury	24.
Pressure in the condenser in pounds per square inch above zero	3.
Number of revolutions made per minute by the screws	372.125
Speed of the screws in geographical miles per hour	30.87916
Slip of the screws in per centum of their axial speed	27.2
Speed of the vessel in geographical miles per hour	22.48003
Speed of the slip of the screws in geographical miles per hour	8.39913
Number of revolutions made per minute by the air pumps and feed pumps.	130.
Number of revolutions made per minute by the circulating pump	360.
Number of revolutions made per minute by the forward blower	500. (?)
Number of revolutions made per minute by the after blower	825. (?)
Height, in inches, of water column equilibrating air pressure in forward fire room	2.5
Height, in inches, of water column equilibrating air pressure in aft fire room	3.0

The consumption of coal as nearly as can be estimated was about 3,900 pounds per hour, or about 50 pounds per hour per square foot of grate. The indicated horse-power was obtained for about 2.223 pounds of coal consumed per hour.

The thrust of the screws in the direction of their axes, calculated from their slip in the manner hereinbefore described, is 14,741.28 pounds, consequently, the indicated horses-power developed by the motive engines, also calculated in the manner hereinbefore described, and its distribution, will be as follows:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engines.....	1,754 4522	
Horses power required to work the engines, *per se*.....	89.3431	
Net horses-power developed by the engines.....	1,665.1091	100.0000
Horses-power absorbed by the friction of the load.....	124.8832	7.5000
Horses-power expended in overcoming the resistance of the water to the surface of the screw blades.....	141.0656	8 4718
Horses-power expended in the slip of the screws.....	380.5716	22.8557
Horses-power expended in the propulsion of the hull..	1,018.5887	61.1725
Totals.....	1.665.1091	100.0000

The vessel's resistance of the above 1,018.5887 horses-power at the experimental speed, was composed of the resistance of the wetted surface of the hull, and of the resistance of the water to displacement. The first can be calculated on the assumption that the resistance of a square foot of wetted surface at the speed of 10 feet per second was 0.45 pound, increasing or decreasing in the ratio of the square of the speed of the surface. The second will, of course, be the remainder after deducting the first from the 1,018 5887 horses-power.

The speed of the surface of the hull, allowance being made for its inclination to the central longitudinal plane of the vessel, was 37.78 feet per second. The area of the wetted surface was 1,831.5 square feet, and its resistance per square foot was 6.423 pounds. Hence the horses-power expended in overcoming the resistance of the wetted surface of the hull were

$$\frac{1{,}831.5 \times 6.423 \times (37.78 \times 60)}{33{,}000} = 808.0609,$$

deducting which from the 1,018.5887, there remain 210.5278 horses-power expended in the displacement of the water alone. Roundly, four-fifths of the power propelling the hull were expended in surface work, and one-fifth was expended in moving bulk of water.

EXPERIMENTS MADE BY A BOARD OF UNITED STATES NAVAL ENGINEERS ON ONE OF THE MOTIVE ENGINES OF THE UNITED STATES TORPEDO BOAT *CUSHING*, IN THE POTOMAC RIVER, TO ASCERTAIN BY MEANS OF THE INDICATOR THE POWER DEVELOPED UNDER THE EXPERIMENTAL CONDITIONS AND THE DISTRIBUTION OF THE PRESSURES IN THE STEAM CYLINDERS.

A Board of United States Naval Engineers consisting of Passed Assistant Engineers H. Webster and C. A. Carr, and Assistant Engineers Emil Theiss, W. H. Chambers and H. G. Leopold, made, by order of the Navy Department, four trials of one of the motive engines of the United States torpedo boat *Cushing*, in the Potomac River, at different speeds of vessel, for the purpose of ascertaining the power developed by the engine under the experimental conditions, and the distribution of the pressure in the steam cylinders. Accordingly, indicators were applied to the starboard engine alone, the port engine not being experimented with, but being operated exactly like the starboard engine, making the same number of double strokes of piston per minute, and having, presumably, the same steam pressures in its cylinders.

The data and results of these trials are given in the following Table No. 4, and there must be kept in mind that they are for one engine and one screw only, though both engines and both screws were in use.

The part of the Potomac River (between the city of Alexandria and Piney Point) in which the trials were made was tide water, but much too shallow and too narrow for accurate experimenting with a vessel of the *Cushing's* dimensions and speed. Further, the vessel was not run forwards and backwards over any measured distance, but was run continuously forward on her route from Alexandria to the City of Norfolk, the speed given being what the commanding officer supposed it to be according to his observations of points along the shore, direction and force

of tide and direction and force of wind. His time for observation, too, was confined within one hour for each trial. Of course, estimations of speed made under such circumstances cannot be accepted for more than a reasonably fair approximation to the truth, but the writer believes them in this case to be very close, and in the following Table No. 4 they have been treated as exact, and the results deduced from them are what would have been under this supposition. The following citation from the report of the Board shows to how great an extent the performance of the vessel was affected by the shoalness of the water:

"The third period, beginning at 3:50 P. M., was, a part of the time, over water too shoal for the best results, the effect being to retard the speed of the boat in some degree, with a corresponding falling off in the engine speed. The character of the 'wake' produced varied with the depth of water over which the boat was passing. During the passage over comparatively shoal water, the water astern assumed an abrupt wall-like character, extending approximately at right angles to the course of the boat, and rising to an estimated height of four feet above the normal level. The normal 'wake' of the *Cushing* in deep water is but slightly above the level of the water, and it spreads away from the stern, forming an outline not unlike that of a ship's plan of a considerable fineness."

During these trials, the mean displacement of the *Cushing* was 125.36 tons, or 20 tons more than during the trials made by Mr. N. G. Herreshoff, Tables Nos. 1 and 2, and during the contract trial made by the Board of Naval Officers. This difference of displacement amounted to 19 per centum more than in the previous trials, and, of course, correspondingly increased the resistance of the hull and the slip of the screws. The increased resistance of the hull, due to its greater immersion, was about one-twelfth. The propelling efficiency of the screws remaining unaltered, their slip would be increased about one twenty-fourth, all other things remaining the same. The trim of the vessel was also very different, being only one-third as much by the stern as in the previous trials referred to.

Each trial was exactly one hour long, during which indicator diagrams were taken from all the cylinders of the starboard engine every 15 minutes and as nearly simultaneous as possible. There were thus four sets of diagrams for each trial. With each set of diagrams a corresponding set of observations of the data was taken.

The number of revolutions made by the screws during the hour was taken with counters, and their mean per minute is given in Table No. 4.

The pressures were taken by gauge, every 15 minutes, of the steam in the boiler, at the valve chest of the small cylinder, and in the 1st, 2d and 3d receivers of the starboard engine. The vacuum in the condenser; the air pressure in the fire rooms; the temperatures of the air in the engine room and fire rooms, and of the feed water and injection water; the revolutions made per minute by the air-pump and feed-pump engines, by the circulating-pump engine, and by the forward and after blowing engines, were noted every 15 minutes. No attempt was made to weigh the coal consumed.

In the following Table No. 4 will be found the observed quantites and those calculated from them, for the four trials of the *Cushing* at different speeds. These quantities are placed in four columns, lettered for reference A, B, C and D. For the same reason the quantities have been grouped and the lines containing them numbered. Each quantity is so completely described on its respective line that but little explanation is needed. During experiments A and B only the after boiler was used. During experiments C and D both boilers were used.

ENGINE.

Lines 1 to 31, both inclusive, contain the data relative to the steam pressures in the boiler, in the steam pipe near the small cylinder, in the 1st, 2d and 3d receivers, and in the condenser, and relative to the conditions under which the steam was used. The boiler pressure, line 2, evidently erroneous in the log of the experiments, is corrected according to the notes of Mr. N. G. Herreshoff.

Table No. 4, containing the data and results of the four trials of the United States torpedo boat *Cushing*, made at different speeds in the Potomac River on the 20th and 21st of July, 1891, by a Board of United States Naval Engineers.

Note: The vessel's mean draught of water amidship during the trials was 5 feet 5 inches, trimming by the stern 11 inches, as measured when at rest. Displacement 125.36 tons. Greatest immersed transverse section 55.4 square feet. Immersed external surface 1984 square feet.

No. of line.		Only the aft boiler in use.		Both boilers in use.	
		A	B	C	D
	ENGINE.				
1	Proportion of throttle valve open	Wide.	Wide.	Wide.	Wide.
2	Steam pressure in boilers in pounds per square inch above the atmosphere	84.0	127.5	203.0	220.0
3	Steam pressure in boilers in pounds per square inch above zero	98.7	142.2	217.7	234.7
4	Steam pressure in steam pipe near valve chest of small cylinder in pounds per square inch above zero	81.0	123.0	196.0	212.0
5	Steam pressure in the receiver between the small cylinder and the first intermediate cylinder, in pounds per square inch above zero	31.0	51.0	90 0	116.0
6	Steam pressure in the receiver between the first and second intermediate cylinders, in pounds per square inch above zero	16.5	24.5	43.5	51.5
7	Steam pressure in the receiver between the second intermediate cylinder and the two large cylinders, in pounds per square inch above zero	7.0	10.0	15.5	21.5
8	Atmospheric pressure, in pounds per square inch above zero	14.7	14.7	14.7	14.7
9	Vacuum in the condenser, in inches of mercury	25 25	24.12	23.00	19.88
10	Back pressure in the condenser, in pounds per square inch above zero	2.306	2.860	3.410	4.942
11	Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.693	0.693	0.693	0.693
12	Fraction completed of the stroke of the piston of the small cylinder when the steam was released	0.936	0.936	0.936	0.936
13	Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned	0 873	0.873	0 873	0 873
14	Fraction completed of the stroke of the piston of the 1st intermediate cylinder when the steam was cut off	0 804	0.804	0.804	0.804
15	Fraction completed of the stroke of the piston of the 1st intermediate cylinder when the steam was released	0.938	0.938	0.938	0.938
16	Fraction completed of the stroke of the piston of the 1st intermediate cylinder when the steam was cushioned	0.841	0.841	0.841	0.841
17	Fraction completed of the stroke of the piston of the 2d intermediate cylinder when the steam was cut off	0.728	0.728	0.728	0.728
18	Fraction completed of the stroke of the piston of the 2d intermediate cylinder when the steam was released	0.953	0.953	0.953	0.953
19	Fraction completed of the stroke of the piston of the 2d intermediate cylinder when the steam was cushioned	0.819	0.819	0.819	0.819
20	Fraction completed of the stroke of the pistons of the two large cylinders when the steam was cut off	0 662	0.662	0.662	0.662
21	Fraction completed of the stroke of the pistons of the two large cylinders when the steam was released	0.944	0.944	0.944	0.944
22	Fraction completed of the stroke of the pistons of the two large cylinders when the steam was cushioned	0.865	0.865	0.865	0.865
23	Number of times the steam was expanded in the small cylinder after the cut off closed	1.3724	1.3724	1.3724	1.3724
24	Number of times the steam was expanded in the engine	10.81847	10.81847	10.81847	10.81847
25	Air pressure above atmosphere, in inches of water column, in fire room of forward boiler	...	...	1.6	2.0

25	Air pressure above atmosphere, in inches of water column, in fire room of forward boiler	10.81847	10.81847	10.81847	10.81847
		...	...	1.6	2.9
26	Air pressure above atmosphere, in inches of water column, in fire room of aft boiler	0.4	1.7	1.2	2.1
27	Number of double strokes made per minute by the steam pistons of the motive engines	205.9	240.0	298.9	328.3
28	Number of double strokes made per minute by the steam pistons of the auxiliary engines working the air pumps and feed pumps	305.	377.	520.	391.
29	Number of double strokes made per minute by the steam piston of the auxiliary engine working the circulating pump supplying the injection water	320.	460.	543.	420.
30	Number of revolutions made per minute by the blower of the forward boiler	...	...	776. (?)	912. (?)
31	Number of revolutions made per minute by the blower of the aft boiler	498. (?)	797. (?)	717. (?)	993. (?)
	TEMPERATURES.				
32	Temperature in degrees Fahrenheit of the air in the engine room	96.	103.	104.	107.
33	Temperature in degrees Fahrenheit of the air in the fire rooms	139.	130.	143.	125.
34	Temperature in degrees Fahrenheit of the injection water	81.	80.	82.	79.
35	Temperature in degrees Fahrenheit of the feed water	88.	104.	126.	149.
36	Temperature in degrees Fahrenheit of the steam considered as saturated	326.6	354.1	388.9	395.4
	SPEED AND SLIP.				
37	Speed of the vessel per hour in geographical miles of 6,086 feet	13 08	13.40	15.42	17.50
38	Axial speed of the screw per hour in geographical miles of 6,086 feet	16 921561	19.915347	24.802905	27.242535
39	Slip of the screw per hour in geographical miles of 6,086 feet	3.841561	6.515347	9.382905	9.742535
40	Slip of the screw in per centum of its speed	22 702167	32.715207	37.829863	35.762219
	STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.				
41	Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero.	78.70	119.50	190.45	206.05
42	Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero.	57.20	85.32	139.45	150.70
43	Pressure on piston of small cylinder at end of its stroke, in pounds per square inch above zero	38.58	59.32	103.45	118.82
44	Mean back pressure against piston of small cylinder, during its stroke, in pounds per square inch above zero.	34.16	53.00	93.30	121.21
45	Mean back pressure against the piston of the small cylinder, exclusive of cushioning, during the remainder of its stroke, in pounds per square inch above zero	33.14	51.44	90.07	116.64
46	Back pressure against the piston of the small cylinder at the point where the cushioning begins, in pounds per square inch above zero	31.45	50.00	86.82	100.57
47	Mean pressure of the expanding steam alone in the small cylinder after the closing of the cut-off valve, in pounds per square inch above zero	48.62	72 53	118.54	128.10
48	Indicated pressure on the piston of the small cylinder, in pounds per square inch	28.58	40.75	60.80	67.60
49	Net pressure on the piston of the small cylinder, in pounds per square inch	24.58	36 75	56.80	63.60
50	Total pressure on the piston of the small cylinder, exclusive of cushioning, in pounds per square inch above zero, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed of the stroke when the steam was cushioned	57 511	85.657	139 431	169.427
	STEAM PRESSURES IN FIRST INTERMEDIATE CYLINDER PER INDICATOR.				
51	Pressure on piston of 1st intermediate cylinder at commencement of its stroke, in pounds per square inch above zero	29.58	48.76	85.26	111.07
52	Pressure on piston of 1st intermediate cylinder at point of cutting off the steam, in pounds per square inch above zero	23.26	31.51	53.01	73.32
53	Pressure on piston of 1st intermediate cylinder at end of its stroke, in pounds per square inch above zero.	20.07	27.32	45.82	62.57
54	Mean back pressure against piston of 1st intermediate cylinder during its stroke, in pounds per square inch above zero	17.00	25.60	44.92	58.07
55	Mean back pressure against the piston of the 1st intermediate cylinder, exclusive of cushioning, during the remainder of its stroke, in pounds per square inch above zero	16.46	24.32	42.70	55.14

Line.	(Table No. 4 continued.)	A	B	C	D
56	Back pressure against the piston of the 1st intermediate cylinder at the point where the cushioning begins, in pounds per square inch above zero	15 57	23.82	41.90	53.95
57	Mean pressure of the expanding steam on the annular surface remaining of the piston of the 1st intermediate cylinder after deducting from the area of that piston the area of the piston of the small cylinder, and during the entire stroke of the piston, in pounds per square inch above zero	26 81	39.04	66.12	89.21
58	Indicated pressure on the piston of the 1st intermediate cylinder, in pounds per square inch	9 81	13.44	21.20	31.14
59	Net pressure on the piston of the 1st intermediate cylinder, in pounds per square inch	7 81	11.44	19.20	29.14
60	Total pressure on the annular surface remaining of the piston of the 1st intermediate cylinder after deducting from the area of that piston the area of the piston of the small cylinder, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed of the stroke when the cushioning of the steam began, in pounds per square inch above zero	23.653	33.893	57.111	77.513
	STEAM PRESSURE IN SECOND INTERMEDIATE CYLINDER PER INDICATOR.				
61	Pressure on piston of 2d intermediate cylinder at commencement of its stroke, in pounds per square inch above zero	15.42	22.95	39.40	49.26
62	Pressure on piston of 2d intermediate cylinder at point of cutting off the steam, in pounds per square inch above zero	8.82	14.26	24.82	32.26
63	Pressure on piston of 2d intermediate cylinder at end of its stroke, in pounds per square inch above zero	7.20	11.64	20.76	26.32
64	Mean back pressure against piston of 2d intermediate cylinder during its stroke, in pounds per square inch above zero	6.30	10.02	18.05	23.64
65	Mean back pressure against the piston of the 2d intermediate cylinder, exclusive of cushioning, during the remainder of its stroke, in pounds per square inch above zero	5.84	9.40	17.06	22.61
66	Back pressure against the piston of the 2d intermediate cylinder at the point where the cushioning begins, in pounds per square inch above zero	5.67	9.39	16.76	22.70
67	Mean pressure of the expanding steam on the annular surface remaining of the piston of the 2d intermediate cylinder after deducting from the area of that piston the area of the piston of the 1st intermediate cylinder, and during the entire stroke of the piston, in pounds per square inch above zero	11.625	17.884	31.27	39.14
68	Indicated pressure on the piston of the 2d intermediate cylinder, in pounds per square inch	5 325	7 864	13.22	15.50
69	Net pressure on the piston of the 2d intermediate cylinder, in pounds per square inch	4.325	6.864	12.22	14.50
70	Total pressure on the annular surface remaining of the piston of the 2d intermediate cylinder after deducting from the area of that piston the area of the piston of the 1st intermediate cylinder, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed of the stroke when the cushioning began, in pounds per square inch above zero	10.108	15.563	27.192	34.018
	STEAM PRESSURES IN THE TWO LARGE CYLINDERS PER INDICATOR.				
71	Pressure on the pistons of the two large cylinders at commencement of their stroke, in pounds per square inch above zero	6.22	9.04	13.90	19.20
72	Pressure on the pistons of the two large cylinders at point of cutting off the steam, in pounds per square inch above zero	4.70	6.40	10.59	14.73
73	Pressure on the pistons of the two large cylinders at end of their stroke, in pounds per square inch above zero	3.73	5.10	7.70	10.90

74	Mean back pressure against the pistons of the two large cylinders during their stroke, in pounds per square inch above zero	3.34	4.13	5.05	7.59
75	Mean back pressure against the pistons of the two large cylinders, exclusive of cushioning, during the remainder of their stroke, in pounds per square inch above zero	3.22	4 00	4.92	7.30
76	Back pressure against the pistons of the two cylinders at the point where the cushioning begins, in pounds per square inch above zero	3.00	3.65	4.44	7.00
77	Mean pressure of the expanding steam on the annular surface remaining of the pistons of the two large cylinders after deducting from the aggregate area of those pistons the area of the pistons of the 2d intermediate cylinder, and during the entire stroke of the pistons, in pounds per square inch above zero	5 308	7.407	11.679	15.864
78	Indicated pressure on the pistons of the two large cylinders, in pounds per square inch	1,968	3.275	6.629	8.274
79	Net pressure on the pistons of the two large cylinders, in pounds per square inch	1.468	2.775	6.129	7.774
80	Total pressure on the annular surface remaining of the pistons of the two large cylinders after deducting from the aggregate area of those pistons the area of the piston of the 2d intermediate cylinder, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed of the stroke when the cushioning began, in pounds per square inch above zero	4.753	6.735	10.885	14.329
	EQUIVALENT PRESSURES ON THE PISTONS OF THE TWO LARGE CYLINDRS PER INDICATOR.				
81	Indicated pressure in pounds per square inch that would be on the pistons of the two large cylinders were the indicated pressures on the piston of the small cylinder and on the pistons of the 1st and 2d intermediate cylinders, divided by the ratio of the respective areas of the pistons of those three cylinders to the aggregate area of the pistons of the two large cylinders, and the quotients added to the experimental indicated pressure on the pistons of the two large cylinders	10.64907	15.65031	26.12609	32.26224
82	Net pressure in pounds per square inch that would be on the pistons of the two large cylinders were the net pressures on the piston of the small cylinder, and on the pistons of the 1st and 2d intermediate cylinders, divided by the ratio of the respective areas of the pistons of those three cylinders to the aggregate area of the pistons of the two large cylinders, and the quotients added to the experimental net pressure on the pistons of the two large cylinders	8.64851	13.64975	24.12553	30.26168
83	Total pressure in pounds per square inch above zero that would be on the pistons of the two large cylinders, corresponding to the total aggregate horses-power developed by the engine	15.04586	22.19119	35.67756	46.54274
84	Back pressure in pounds per square inch above zero that would be against the pistons of the two large cylinders, corresponding to their experimental back pressure plus the difference in pounds per square inch between the back pressure, exclusive of cushioning, against the piston of the small cylinder and the mean steam pressure on the piston of the 1st intermediate cylinder from the commencement of its stroke to the point of cutting off the steam, and between the back pressure, exclusive of cushioning, against the piston of the 1st intermediate cylinder and the mean steam pressure on the piston of the 2d intermediate cylinder from the commencement of its stroke to the point of cutting off the steam, and between the back pressure, exclusive of cushioning, against the piston of the 2d intermediate cylinder and the mean steam pressure on the pistons of the two large cylinders from the commencement of their stroke to the point of cutting off the steam, all divided by the ratios which the areas of the respective pistons have to the aggregate area of the pistons of the two large cylinders	4.39679	6.54088	9 55147	14.28050
	EXPANDING PRESSURES COMPARED.				
85	Mean pressure of the expanding steam, in pounds per square inch above zero, calculated according to the Mariotte law	11.47517	17.11645	27.99579	30.23265
86	Experimental mean pressure of the expanding steam, in pounds per square inch above zero	11.02324	16.17546	27.09441	35.08712

Line.	(Table No. 4 continued.)	A	B	C	D
	HORSES-POWER.				
87	Indicated horses-power developed in the small cylinder	43.777798	72.753939	135.190804	165.095500
88	Net horses power developed in the small cylinder	37.650744	65.612448	126.296672	155.326535
89	Total horses-power developed in the small cylinder	88.093245	152.929672	310 029424	413.781586
90	Horses-power developed by the expanding steam alone in the small cylinder	22.863621	39.754377	80.918322	96.045284
91	Indicated horses-power developed in the 1st intermediate cylinder	30.582774	48.838468	95.943000	154.789326
92	Net horses-power developed in the 1st intermediate cylinder	24.347754	41.570839	86.891774	144.847815
93	Total horses-power developed in the 1st intermediate cylinder	37.507665	62.646774	131.468933	195.985024
94	Horses-power developed by the expanding steam alone in the 1st intermediate cylinder	42.513868	72.160330	152.207558	225.559893
95	Indicated horses power developed in the 2d intermediate cylinder	32.895677	56.626226	118.555101	152.674130
96	Net horses-power developed in the 2d intermediate cylinder	26.718085	49.425536	109.587241	142.824186
97	Total horses-power developed in the 2d intermediate cylinder	30.931308	55.511283	120.789207	165.980231
98	Horses-power developed by the expanding steam alone in the 2d intermediate cylinder	35.573452	63.790001	138.904034	190.971434
99	Indicated horses-power developed in the two large cylinders	24 315002	47.164519	118.895879	162.996871
100	Net horses-power developed in the two large cylinders	18.137410	39.963829	109.928020	153.146928
101	Total horses-power developed in the two large cylinders	29.362095	48.496032	97.615149	141.139846
102	Horses-power developed by the expanding steam alone in the two large cylinders	32.790658	53.334015	104.735630	154.259510
103	Aggregate indicated horses-power developed by the engine	131.571251	225.383152	468.584784	635.555827
104	Aggregate net horses power developed by the engine	106.853993	196.572652	432.703707	596.145464
105	Aggregate total horses-power developed by the engine	185.894313	319.583761	659 902713	916.886687
106	Aggregate horses power developed by the engine by the expanding steam alone	133.741599	229.039323	476.765544	666.836081
	DISTRIBUTION OF THE INDICATED HORSES-POWER.				
107	Horses-power expended in working the engine *per se*, or unloaded	24.717258	28.810500	35.881077	39.410363
108	Horses-power absorbed by the friction of the load	8.014049	14.742949	32.452778	44.719910
109	Horses-power expended in overcoming the cohesive resistance of the water to the helicoidal surface of the blades of the screw (both sides)	11.951425	18.921600	36.558557	48.441340
110	Horses-power expended in the slip of the screw	19.725577	53.295723	137.584326	179.881535
111	Horses-power expended in the propulsion of the vessel	67 162942	109.612380	226.108046	323.111679
	DIVISION OF POWER APPLIED TO HULL.				
112	Horses-power expended in overcoming the cohesive resistance of the water to the immersed external or wetted surface of the hull	54.543450	58.695250	89.366000	130.677100
113	Horses-power expended in simply displacing the water, irrespective of the wetted surface	12.619492	50.917130	136.742046	192.434579
114	Thrust of the screw in pounds, calculated from the above distribution of the power	1670 5336	2661.2646	4770.5108	6006.8647
	RATIOS.				
115	Ratio of the equivalent net pressures on the pistons of the two large steam cylinders	1.000000	1.578278	2.789559	3.499063
116	Ratio of the thrusts of the screw: also ratio of the experimental resistances of the hull	1.000000	1 593063	2.855681	3.595776
117	Ratio of the square roots of the thrust of the screw	1.000000	1.262166	1.689876	1.896253
118	Ratio of the slips of the screw in geographical miles per hour	1.000000	1.696016	2.442472	2.536087
119	Relative resisting efficiencies of the screw surface	1.000000	0.553826	0.478607	0.559068
120	Relative resistances of the hull according to the law of the square of the speed	1.000000	1.075206	1.638442	2.394918

120	Relative resistances of the hull according to the law of the square of the speed	1.000000	1.075206	1.638442	2.394918
121	Per centum by which the experimental resistances of the hull exceed the resistances according to the law of the square of the speed	0.000000	48.163515	74.29246	50.141926
	WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.				
122	Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there	1988.579	3303.684	6264.022	7228.137
123	Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there	1857.741	3224.305	6688.542	8294.891
124	Number of pounds of steam liquefied per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in that cylinder	62.492	111.023	232.045	277.110
125	Sum of the two preceding quantities	1920.233	3335.328	6920.587	8572.001
126	Number of pounds of steam present per hour in the 1st intermediate cylinder at the end of the stroke of its piston, calculated from the pressure there	1983 644	3018.355	6073.459	9078.918
127	Number of pounds of steam liquefied per hour in the small and in the 1st intermediate cylinders to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut off on the small cylinder	175.591	306.211	654.638	913.798
128	Sum of the two immediately preceding quantities	2159 235	3324.566	6728.097	9992.716
129	Number of pounds of steam present per hour in the 2d intermediate cylinder at the end of the stroke of its piston, calculated from the pressure there	1432.545	2625.500	5703.296	7729.725
130	Number of pounds of steam liquefied per hour in the small cylinder and in the 1st and 2d intermediate cylinders to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut off on the small cylinder	267.227	473.157	1026.612	1430.420
131	Sum of the two immediately preceding quantities	1699.772	3098.657	6729.908	9160.145
132	Number of pounds of steam present per hour in the two large cylinders at the end of the stroke of their pistons, calculated from the pressure there	1609.535	2559.747	4862.862	7329.134
133	Number of pounds of steam liquefied per hour in the small cylinder, in the 1st and 2d intermediate cylinders, and in the two large cylinders, to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut off on the small cylinder	349.577	608.483	1296.426	1832.301
134	Sum of the two immediately preceding quantities	1959.112	3168.230	6159.288	9161.435
	ESTIMATED WEIGHT OF STEAM CONSUMED PER HOUR.				
135	Estimated number of pounds weight of steam consumed per hour by the engine, on the supposition that this number was the maximum quantity of steam shown in each case in the above group of the weight of steam accounted for by the indicator by calculation from the pressure at the point of cutting off the steam or at the end of the stroke of the piston added to the weight of steam liquefied to furnish the heat transmuted into the aggregate total horses-power developed in all the cylinders by the expanding steam alone (line 133), and the sum increased one-sixth	2727.849	4564.195	9315.796	12729.755
	ECONOMIC RESULT.				
136	Number of pounds of steam consumed per hour per indicated horse-power	20.733	20.251	19.881	20.029
137	Number of pounds of steam consumed per hour per net horse-power	25.529	23.219	21.529	21.353
138	Number of pounds of steam consumed per hour per total horse-power	14.674	14.281	14.117	13.884

The point at which the steam was cut off in the different cylinders, lines 11, 14, 17 and 20, is the point on the indicator diagrams where the convex curve of the throttled steam changes into the concave curve of the expanded steam. The point at which the steam is released from the different cylinders, lines 12, 15, 18 and 21, is the point at which the concave expansion curve changes into the convex curve of the escaping steam. The point at which the steam is cushioned in the different cylinders, lines 13, 16, 19 and 22, is the point where the compression curve of the back pressure leaves the straight line of the back pressure tangentially. Of course, all of these points are determined by the eye, and cannot be determined in any other manner, on account of the disturbance caused by the various expansions of the metal due to the various temperatures of the mechanism when in use, to the play of the journals of the valve gear under its various strains and to inaccuracies of workmanship.

Line 23 gives the number of times the steam was expanded in the small cylinder. This is the quotient of the division of the sum of the waste space at one end of the cylinder and of the space displacement of the piston per stroke by the sum of the waste space at one end of the cylinder and of the space displacement of the piston per stroke up to the point of cutting off the steam.

Line 24 gives the number of times the steam is expanded in the engine. It is the quantity on line 23 multiplied by the quotient of the division of the sum of the waste spaces at one end of the two large cylinders and of the space displacements of their pistons per stroke, by the sum of the waste space at one end of the small cylinder and of the space displacement of its piston per stroke. Of course, the same result would be obtained by dividing the sum of the waste space at one end of the small cylinder and of the space displacement of its piston per stroke up to the point of cutting off, into the sum of the waste spaces at one end of the two large cylinders, and of the displacement of their pistons per stroke. In this connection the two large cylinders are considered as one cylinder of their aggregate capacity.

What is thus called the number of times the steam is ex-

panded, merely expresses the ratio of the space in the small cylinder between its valve and piston when the steam was cut off, to the space in the two large cylinders between their valves and pistons when the latter are at the end of their stroke. The ratio of the expansion of the steam, however, may be quite different from this, but cannot be ascertained owing to the various unknown liquefactions of the steam and vaporizations of the water of these liquefactions in the cylinders. Any steam leakage into or out of the cylinders likewise affects the ratio of expansion. Furthermore, this ratio of expansion does not include the expansion which the steam undergoes in the small cylinder from the initial pressure there to the pressure at the point of cutting off. In fact, from the moment when the steam leaves the boiler by entering the steam pipe, it undergoes a continuous expansion in the latter; and from the moment it enters the small cylinder, it undergoes continuous expansion both before as well as after the closing of the cut off valve; and from the moment it leaves the small cylinder it undergoes continuous expansion until it is annihilated in the condenser. The closing of cut off valves on intermediate cylinders does not in the slightest degree affect the expansion. The steam expands without interruption from its leaving the boiler to its arrival into the condenser. From one end to the other of this route there is a continuous decrease of pressure, and an aeriform body can only have its pressure decreased by enlargement of its volume, which is expansion. Conversely, such a body can only have its pressure increased by diminution of its volume, which is compression. The steam flows from the boiler to the condenser solely by continuous difference of pressure. At each successive point of the route the pressure must be less than at the immediately preceding point. Moreover, the expanding steam, from the moment it commences its movement from the boiler until its annihilation in the condenser, does work, either mass, or molecular, or both, and necessarily undergoes a liquefaction of such portion as may be required to liberate the heat transmuted into such work. The steam, therefore, when it enters the small cylinder is always wet steam, although it may be perfectly dry steam in the boiler, that

is to say, it is laden with the infinitely small particles of water of liquefaction equally dispersed among it, due to the performance of work, and the wetness continues to increase the more the steam is expanded.

The release of the steam from the cylinder before the stroke of the piston is completed is not wholly a loss, inasmuch as it allows the piston to commence its return stroke with less back pressure against it than it would have had the steam been retained to the end of the stroke.

The cushioning is an economic gain thermally, to the extent that the cylinder surfaces may have absorbed the heat of compression. Dynamically, there is neither gain nor loss; whatever work is done in producing the pressure of the compressed steam is returned upon the piston during the next stroke.

The number of double strokes made per minute by the pistons of the steam cylinders working the air pumps and feed pumps (line 28), working the circulating pump (line 29), and working the two fan-blowers (lines 30 and 31), cannot be depended on as exact, not having been taken continuously by counters, but only at intervals by counting with a watch; and the same remark applies to these quantities in the preceding experiments.

The vacuum in the condenser (line 9), at the higher speeds of vessel (columns C and D), was poor, but accompanied by high temperature of feed water (line 36).

The difference between the boiler pressures (line 3) and the pressures in the steam pipe near the valve chest of the small cylinder (line 4) are very great, being, in pounds per square inch, 17.7 for experiment A, 19.2 for experiment B, 21.7 for experiment C, and 22.7 for experiment D, increasing with increased velocity of piston and increased pressure of steam. Other things equal, the greater the velocity of the piston the greater will be this difference, and the greater the steam pressure the greater will be this difference also, and, for the latter quantity, in the inverse ratio of the square root of the density of the steam, which is approximately represented by the pressure above zero.

STEAM PRESSURES ON THE PISTONS OF THE DIFFERENT CYLINDERS.

Lines 41 to 80, both inclusive, show the distribution of the pressure on the steam pistons of the different cylinders.

The difference in the small cylinder between the initial pressure and the pressure at the point of cutting off the steam (lines 41 and 42,) is very great, being, in pounds per square inch, 21.5 for experiment A, 34.18 for experiment B, 51 for experiment C, and 55.35 for experiment D. These figures show a very large expansion of the steam in the cylinder previous to the closing of the cut-off, just as the large difference between the boiler pressure and the pressure in the steam pipe adjacent the valve chest of the small cylinder, shows a similar considerable expansion in the steam pipe.

The difference in pounds per square inch between the pressure in the boiler (line 3) and the pressure in the small cylinder at the point of cutting off the steam (line 42), is 41.50 for experiment A, 56.88 for experiment B, 78.25 for experiment C, and 84.00 for experiment D. Or, taking the boiler pressure in each case as unity, the degradation of pressure was 42.05 per centum for experiment A, 40.00 per centum for experiment B, 35.94 per centum for experiment C, and 35.75 per centum for experiment D. Of course, the expansion curves due to these great differences of pressure would not be like the expansion curves produced by cutting off the steam, in which the same weight of steam is expanded throughout, because, in the steam pipe, and in the small cylinder previous to cutting off the steam, the steam is simultaneously expanding and receiving fresh accessions of weight.

The pressures at the end of the stroke of the piston of the small cylinder, line 43, are the following per centum of the pressures at the point of cutting off the steam, line 42, namely:

For experiment A, ...	67.45
For experiment B, ...	69.52
For experiment C, ...	74.18
For experiment D, ...	78.84

showing that the steam liquefactions in the small cylinder, due to all causes, were less with the greater piston speeds and

pressures than with the smaller ones. Nevertheless, assuming the pressures at the point of cutting off the steam in the small cylinder, line 42, to represent, relatively, the weight of steam received from the boiler by that cylinder in the different experiments, and comparing with them the mean total pressures, line 50, produced on the piston by such weights, there will be found a tolerably close equality for all the experiments, namely:

	Cut-off pressures.	Mean total pressures.
Experiment A, . . .	57.20	57.511
Experiment B, . .	85.32	85.657
Experiment C, . . .	139.45	139.431
Experiment D, . .	150.70	169.427

In the case of the 1st intermediate cylinder, the pressures at the end of the stroke of the piston, line 53, are very nearly the same per centum of the pressures at the point of cutting off the steam, line 52, for all the experiments, namely:

For experiment A, ...	86.28
For experiment B, . . .	86.70
For experiment C, ...	86.44
For experiment D, . . .	85.34

Assuming the pressures at the point of cutting off the steam in the 1st intermediate cylinder, line 52, to represent, relatively, the weight of steam received by that cylinder in the different experiments, and comparing with them the mean total pressures, line 60, produced on the piston by such weights, there will be found the following for the different experiments, namely:

	Cut-off pressures.	Mean total pressures.	Mean total pressures divided by cut-off pressures.
Experiment A,	23.26	23.653	1.017
Experiment B,	31.51	33.893	1.075
Experiment C,	53.01	57.111	1.077
Experiment D,	73.32	77.513	1.072

showing a very close approach to equality of economic effect.

In the case of the 2d intermediate cylinder, the pressures at

the end of the stroke of the piston, line 63, are very nearly the same per centum of the pressures at the point of cutting off the steam, line 62, for all the experiments, namely:

For experiment A, ...	81.63
For experiment B, . . .	81.63
For experiment C, ...	83.64
For experiment D, . . .	81.59

Assuming the pressures at the point of cutting off the steam in the 2d intermediate cylinder, line 62, to represent, relatively, the weight of steam received by that cylinder in the different experiments, and comparing with them the mean total pressures, line 70, produced on the piston by such weights, there will be found the following for the different experiments, namely:

	Cut-off pressures.	Mean total pressures.	Mean total pressures divided by cut-off pressures.
Experiment A, . .	8.82	10.108	1.146
Experiment B, .	14.26	15.563	1.091
Experiment C, . .	24.82	27.192	1.095
Experiment D, .	32.26	34.018	1.054

showing a close approach to equality of economic effect.

In the case of the two large cylinders, the pressures at the end of the stroke of their pistons, line 73, are very nearly the same per centum of the pressures at the point of cutting off the steam, line 72, for all the experiments, viz:

For experiment A, ...	79.36
For experiment B, . . .	79.69
For experiment C, ...	72.71
For experiment D, . . .	74.00

Assuming the pressures at the point of cutting off the steam in the two large cylinders, line 72, to represent, relatively, the weight of steam received by those cylinders in the different experiments, and comparing with them the mean total pressures, line 80, produced on the pistons by such weights, there will be found for the different experiments, namely:

	Cut-off pressures.	Mean total pressures.	Mean total pressures divided by cut off pressures.
Experiment A, . .	4.70	4.753	1.011
Experiment B, .	6.40	6.735	1.052
Experiment C, . .	10.59	10.885	1.028
Experiment D, .	14.73	14.329	0.973

showing a close approach to equality of economic effect.

There must here be remembered that the steam from the auxiliary engines was exhausted into the receiver from which the two large cylinders drew their steam; but, as the power developed by the auxiliary engines must have been nearly proportional to that developed by the main engines, the *relative* results could have been but little affected.

Lines 85 and 86 compare the mean pressure of the expanding steam as given by the Mariotte law, and as experimentally obtained. Of course, the exhaust steam from the auxiliary engines delivered into the 3d receiver vitiate the comparison to that extent. The disturbance thus produced, however, is small, and is in the direction of increasing the experimental pressures, line 86, so that these pressures, excluding the disturbance, would be still less than they appear. Furthermore, the steam undergoes expansion in the waste spaces at the ends of all the cylinders, and the pressure due to such expansion is not included in the experimental pressures on line 86. Could it have been included it would have increased those pressures in a notable degree, so that the final result would doubtless have been that the mean pressure of the expanding steam as given by the Mariotte law did not sensibly differ from the experimental mean pressure.

The important fact in comparisons of the simple or single cylinder engine with compound or triple or more expansion engines, better described as multiple-cylinder engines, is that, for all kinds of engines with cylinders not too much below the medium size, there is really an amazingly close agreement between the mean pressure of the expanding steam according to the Mariotte law and according to the indicator diagram. This coincidence—purely accidental—allows the loss of effective pressure to be ascertained for the multiple cylinder engines due to

expanding the steam in two or more cylinders instead of expanding it the same number of times in one cylinder only. All that is required is to calculate, according to the Mariotte law, from the waste space and the pressure at the point of cutting off the steam in the small cylinder, the mean pressure due to the number of times the steam is expanded in the multiple-cylinder engine, and subtract from this mean pressure the back pressure against the piston of the large cylinder; the remainder will be the indicated or effective pressure. Then from the indicator diagrams of the multiple-cylinder engine ascertain by measurement the mean indicated pressure for all the cylinders, measuring that pressure up to a horizontal line passing through the point of cutting off in the small cylinder, and deduct this measured mean indicated pressure from the calculated mean indicated pressure according to the Mariotte law, and the difference will be the loss of effective pressure due to the transferrence of the steam from cylinder to cylinder of the multiple-cylinder engine. There will always be found a very considerable difference in these two mean pressures, the multiple-cylinder engine giving a much less measured indicated or effective pressure than the calculated indicated pressure according to the Mariotte law.

To correctly compare the single-cylinder with the multiple-cylinder engine in respect of mean pressure produced by the same weight of steam of the same pressure, used with the same measure of expansion, there must be assumed for both the same fraction of the space displacement of the piston in waste space at the ends of the single cylinder of the simple engine, and at the end of the small cylinder of the multiple-cylinder engine. Now, as the equivalent single-cylinder engine will have the same-sized cylinder as the large cylinder of the multiple-cylinder engine, evidently more steam will be required to fill the waste space of the former, in the proportion of the capacity of the large cylinder to that of the small cylinder; consequently, to obtain the same measure of expansion for the steam in the large cylinder of the simple engine, it will have to be cut off earlier in the stroke of the piston. Compare, then, the mean indicated pressure for the single-cylinder engine calculated ac-

cording to the Mariotte law for the waste space and measure of expansion in that engine, with the mean indicated pressure obtained from the multiple-cylinder engine calculated according to the Mariotte law for the waste space in the small cylinder of that engine, for the same measure of expansion and for the same back pressure against the piston of the large cylinder of the multiple-cylinder engine, as against the piston of the single-cylinder engine.

In calculating the mean pressure of the expanding steam for the multiple-cylinder engine, care must be taken to include as space into which the steam is expanded the waste spaces at the ends of the cylinders as well as the spaces displaced by the pistons. But of the pressure produced by the expanding steam, only the portions which fall within the spaces displaced by the pistons are utilized as effective, the pressures in the waste spaces being entirely lost in that respect. The mean pressures thus obtained for the single-cylinder engine and for the multiple cylinder engine will show their relative economic merits, supposing the cylinder liquefactions of steam and the vaporizations of the water of such liquefactions to be the same in both cases. They will show the economic effects of the less steam drawn from the boiler to fill the waste space at the end of the small cylinder of the multiple-cylinder engine, and at the end of the single cylinder of the equivalent simple engine, combined with the economic effects of the waste spaces of the cylinders of the multiple-cylinder engine other than the waste space of the small cylinder, through which the steam is expanded without producing effective work, and combined with the loss of pressure caused by the transferrence of the steam from one cylinder to another of the multiple-cylinder engine.

From the foregoing will be seen that the waste spaces of the cylinders of the multiple-cylinder engine, other than the waste space of the small cylinder, operate a very considerable economic loss. The effective work lost by such waste spaces is represented by the product of the pressure of the steam expanding in them and their capacities, for, evidently, were there no such waste spaces, the pistons of the respective cylinders could be given a

proportionally longer stroke and would develop a correspondingly greater power with exactly the same quantity of steam expended. As regards economy of steam, therefore, it is of quite as much importance to diminish the waste spaces of the cylinders of a multiple-cylinder engine, other than the waste space of the small cylinder, as to diminish the waste space of that cylinder.

The capacities of the receivers between the different cylinders of a multiple-cylinder engine have a great effect on the economy of the steam. What is the receiver capacity that will produce the maximum economic effect? It is the capacity so great that the steam pressure in the receiver will remain sensibly constant during a revolution of the crank shaft.

The essential disadvantage of a receiver is, that it causes the back pressure against the piston of the cylinder discharging into it to be greater than the steam pressure between the commencement of the stroke of the piston and the point of cutting off in the cylinder drawing steam from the receiver. This difference of pressure is pure loss of economic effect, being simply back pressure overcome, that is, ineffective work done; and good designing consists in so proportioning the receiver as to reduce this back pressure to a minimum. To accomplish this, the pressure must remain constant in the receiver—must not fluctuate during a revolution of the crank shaft—so that the difference of the above two pressures will be only what is necessary to force the steam through the valve openings and passages of the two cylinders, which may be taken at about 2 pounds per square inch. Now the strokes of the pistons of the cylinders of a receiver multiple-cylinder engine do not coincide, that is to say, one cylinder does not receive steam directly from its predecessor, beginning its stroke of piston as the other finishes its stroke. On the contrary, the preceding cylinder discharges its steam into the receiver, or finishes its stroke of piston, considerably before the following cylinder commences its stroke of piston, or begins to draw the steam from the receiver. This causes the back pressure against the piston of the preceding cylinder above the steam pressure of the following cylinder, as above explained, to be a max-

imum. Now when the following cylinder commences its draught of steam from the fixed quantity in the receiver, the steam pressure in that cylinder will be less and less as the capacity of the receiver is smaller and smaller, the result being that the difference between the back pressure in the preceding cylinder and the steam pressure in the succeeding cylinder between the commencement of the stroke of its piston and the point of cutting off, if it have a cut-off, becomes much greater than would be were the receiver capacity large enough to prevent sensible variations of pressure in it. The loss of economic effect due to too small a receiver may be very large; but this evil may be remedied within certain limits by cutting off the steam in the following cylinder so as to reduce the *bulk* of a given weight of steam taken out of the receiver thereby less affecting the pressure therein. This *relatively* enlarges the receiver, just the same as throttling the steam in the steam pipe, or cutting it off shorter in the cylinder *relatively* increases the steam room in the boiler, the same *weight* of steam being drawn from the boiler in equal times in both cases, the question being of relative *bulks* only.

The beneficial effect of cutting off the steam in those cylinders of a multiple-cylinder engine which follow the small cylinder, consists principally of this consequent lessening, as just described, of the difference between the back pressure in the preceding cylinder and the steam pressure in the following cylinder, and the cutting off should be so much the shorter as the receiver is smaller. With a sufficiently large receiver, scarcely any benefit would result from cut-offs on the following cylinders. With the tandem arrangement of cylinders, neither receivers between the cylinders, nor cut-offs on the following cylinders, are necessary. Of the two types of multiple-cylinder engines—the tandem and the receiver—both are equally economical provided (other things being equal) the difference between the back pressures of the preceding cylinders, and the steam pressures prior to the closing of the cut-off on the following cylinders, are equal in both, an equality that can only be obtained in practice by the use of as large receivers as practicable, combined with cut-offs as short as practicable.

Cut-offs on the following cylinders, independently of their effect in relatively enlarging the receivers, may produce economy by using the steam expansively which results from the vaporization of the water due to the liquefaction of steam in the preceding cylinders. This water is boiled off in those cylinders as in boilers, by the heat of the metal and under the reduced pressures of the expanding steam.

The foregoing are very important points, which the writer believes have never before been investigated.

The economic performance of the *Cushing's* receiver multiple-cylinder engines might doubtless have been improved by the use of a small cylinder of greater capacity combined with a shorter cut-off and by the use of larger receivers combined with shorter cut-offs on the following cylinders. The weight of the engines, *per se*, and their bulk, would have been thereby increased, but, for equal powers developed, the weight and bulk of the boiler would have been diminished, and the weight of coal carried in the bunkers for steaming equal distances at equal speeds would have been lessened. Of course, the cost of the difference of fuel saved would have been an entire gain.

An increased economy of, say, one-eighth, in the production of the power, means that much decrease in the weight and bulk of boilers and coal bunkers, a very serious matter when their large weights and bulks in a vessel are considered. Economy in the production of the power is as necessary for obtaining maximum power from a given weight and bulk of machinery and of coal for a given time, as for effecting money saving in the cost of fuel.

THRUST OF THE SCREW.

Evidently, the thrust of the screw can only be calculated from its slip, as herein precedingly described, in the cases where the water gets "solidly" upon the whole of the helicoidal surface, which can only happen when the water is of sufficient depth below the vessel's keel to produce that effect. When the depth is insufficient, the water cannot rise vertically at the stern of the vessel in time to refill the void caused by its passage, and cannot, consequently, rise fast enough to get "solidly" upon the

whole of the helicoidal surface. The result is that only a portion of that surface comes propulsively into use, and the slip of the screw is correspondingly increased. There is, therefore, a twofold injurious effect produced by shoal water below a vessel. First, the resistance of the hull, *per se*, at a given speed, is greatly increased; and, second, the slip of the screw is greatly increased above what is due to this increased resistance of the hull. These facts are evidenced by the greater "squatting," for equal speeds, of the hull in too shoal water than in sufficient depth of water, and by the "wild" steering of the vessel due to the facts of the water not getting "solidly" upon the whole of the rudder, and of its getting "solidly" on only the helicoidal surface in the lower portion of the screw's disc. In the latter case, the obliquity of the screw surface is entirely unbalanced, and the action of the screw tends to strongly turn the vessel from its course. The greater the speed of a given vessel and the less the depth of water beneath it—after a certain limit is reached—the greater is the resistance of the hull, the greater is the slip of the screw due to the greater resistance of the vessel and the lesser acting surface of the screw, the greater is the "squatting" of the hull, and the "wilder" is the steering due to the smaller acting surface of the rudder, and to the uncounteracted obliquity of the portion of the screw surface which does act propulsively. If the void created by the passage of the vessel through water was refilled by water flowing in horizontally instead of vertically, too shoal water would not produce any of the above effects, and the undisputed and indisputable fact that such effects are produced, is one of the many proofs that the water of replacement at the stern of a vessel comes vertically from beneath. The injurious consequences of too shoal water thus become more and more marked as the speed of the vessel increases and as the depth of the water beneath the vessel decreases. A depth which is not injurious at one speed may become injurious at a greater speed. The force of these considerations must be appreciated in order to understand the results given by the experiments with the *Cushing* in the Potomac River, where they were enormously influenced by the high speed of the vessel and the low and var-

iable depth of the water. These experiments strikingly illustrate the above principles. In cases, therefore, where there is too little depth of water beneath the vessel, the thrust of the screw must be obtained from the distribution of the indicated horses-power developed by the engine.

The thrusts of *one* screw, line 114, obtained from the distribution of the indicated horses-power, and the thrusts of one screw calculated from its slip in geographical miles per hour, as given on line 39, are as follows, and from them the per centum of the helicoidal surface, which was in each experiment utilized propulsively, are calculated by multiplying the former by 100 and dividing the products by the latter, the quotients being the quantities in the last column:

	Thrust of the screw in pounds obtained from the distribution of the indicated horses-power, line 114.	Thrust of the screw in pounds calculated as hereinbefore described.	Per centum of the helicoidal surface utililized propulsively.
Experiment A,	1,670.5336	1,541.90	100.0000
Experiment B,	2,661.2646	4,435.20	60.0333
Experiment C,	4,770.5108	9,198.35	51.8627
Experiment D,	6,006.8647	9,916.95	60.5717

The speed of the vessel in experiment A, line 37, is given very nearly correct, as the slip of the screw, line 40, is 22.702167 per centum of its axial speed, which is about what it should have been. Assuming then the speed of the vessel in experiment A to be correct, and that the water in the part of the river where the experiment was made was not too shoal for accurate results, the difference of the two determinations of the thrust of the screw, $\left(\frac{1670.5336 - 1541.90 \times 100}{1541.90} =\right)$ 8.34254 per centum of the calculated thrust, is not too large, especially when the uncertainties of the data from which the thrust of the screw was obtained in the distribution of the indicated horses-power are considered. If the thrust of the screw obtained from the distribution of the power is assumed to be the above 8.34254 per centum too large, then the per centum of the helicoidal surface utilized, as shown

in the last column above, will be as follows, and is very nearly accurate:

	Thrust of the screw in pounds, obtained from the distribution of the indicated horses-power, line 114, divided by 1.0834254.	Thrust of the screw in pounds calculated as hereinbefore described.	Per centum of the helicoidal surface utilized propulsively.
Experiment A,	1,541.90	1,541.90	100.0000
Experiment B,	2,456.34	4,435.20	55.3829
Experiment C,	4,403.17	9,198.35	47.8691
Experiment D,	5,544.33	9,916.95	55.9076

The quantities in the last column immediatly above, correspond exactly with those on line 119 calculated by an entirely different process, and on wholly different assumptions.

The quantities on line 119 are the fractions of the helicoidal surface utilized propulsively in the different experiments, and are calculated purely relatively from the ratio of the square roots of the thrust of the screw, line 117, the thrusts in pounds being those on line 114, and from the ratio of the slips of the screw in geographical miles per hour, line 118, the slips being those on line 39, as follows: The quantities on line 118 are divided respectively by those on line 117, the quotient squared, and the squares divided into unity or 1 respectively, the resulting quotients being the quantities on line 119. As the results are entirely relative no corrections are needed in this calculation of the thrusts of the screw on line 114.

Had all the experiments investigated in this paper been more complete, so as to furnish numerical data on all the questions involved, the results would have been not only more satisfactory but more varied, as many points of interest must now be left unnoticed. When experiments are being made, the labor is but little more to observe and record all the data than to obtain only that which is needed for the immediate purpose in view. Generally, however, there will be found that even this purpose cannot be answered with certainty without considerable approach to completeness of data. One great value of experiments is in relation to the general laws of mechanical philosophy applied to

physical research that they may be made to illustrate or confirm, and they are worthy of attention not so much for what they furnish in their particular case as for the general inferences they enable to be drawn applicable to other cases. Unless, also, the experimental results conform to the laws of mechanical philosophy, there will always be great reason to doubt their accuracy, and the data should be complete enough to allow the comparison to be made.

II.

TREATMENT OF NON-ALGEBRAIC CURVES FOR MAXIMA AND MINIMA BY THE USE OF ORDINATES.

By Prof. Wm. F. Durand, of Cornell University, Member.

In engineering investigation the following problem occasionally presents itself:

Given a series of values of a function whose form is unknown, or which may be assumed to be non-algebraic in character. The function is known to pass through a maximum or minimum value. This may be seen from the series of values known, or it may follow from the conditions of the problem. It is required to locate the maximum or minimum values.

The solution must of course be approximate. There are two methods open. 1st. To plat the known values in the neighborhood of the maximum or minimum, draw the curve, and locate the desired points by geometrical means. 2d. To express the desired location as a function of the values in its neighborhood, by a method analagous to that used in Simpson's and other similar methods of approximate integration.

We will treat of the latter method first. To this end we replace the given function in the neighborhood of the maximum value, for instance, by an algebraic function of the form

$$y = ax^n + bx^{n-1} + cx^{n-2} + \quad . \quad . \quad . \quad px + q.$$

For simplicity we will assume it of the second degree only, when we shall have

$$y = ax^2 + bx + c,$$

$$\text{whence } \frac{dy}{dx} = 2ax + b.$$

The condition for y maximum is therefore $x = -\frac{b}{2a}$.

Take now three successive values of the function containing the maximum value between the extremes, and for present purposes consider the axis of y as the mean ordinate. In terms of the common interval as unit, the abscissæ of the other ordinates will be -1 and $+1$.

The corresponding values of the assumed functions in terms of the same units are

$$y_1 = a - b + c,$$
$$y_2 = \quad\quad c,$$
$$y_3 = a + b + c.$$

$$\text{Whence } \frac{y_3 - y_1}{2} = b,$$

$$\text{and } y_1 + y_3 - 2y_2 = 2a,$$

and therefore $x = \frac{y_3 - y_1}{4y_2 - 2y_1 - 2y_3}$, gives the location of the maximum value, relative to the middle ordinate of the three used.

In practice the following form of x is found convenient:

$$x = \frac{y_3 - y_1}{2((y_2 - y_1) + (y_2 - y_3))}.$$

By assuming the algebraic function of the third degree, a value of x involving four ordinates may be found, but vastly more complex in form.

The above value is simple in form, readily used, and if the successive values are sufficiently near, to in any manner correctly represent the function, the result will be quite accurate enough for all ordinary purposes.

If the two extreme ordinates are equal, we find as we should expect that $x = o$. That is, the maximum ordinate lies midway between two closely neighboring ordinates of equal length.

This suggests the geometrical method which is so simple as to be self-apparent. It may however be summarized as follows:

Draw parallel to the axis of abscissæ a chord, the shortest possible which shall make fairly distinct intersection with the curve. Bisect the chord, and the maximum ordinate is thus located.

The geometrical method is, of course, readily apparent, and it is the object of the present note to draw attention rather to the method by ordinates, whereby the conditions for maximum or minimum values are readily found from a few values of the function in their neighborhood, without the necessity of a geometrical construction of the corresponding curve.

III.

A STUDY OF THE ELEMENT OF A SCREW PROPELLER.

By Prof. Wm. F. Durand, of Cornell University, Member.

In 1878, the late Wm. Froude presented to the Institute of Naval Architects a paper in which he discussed the efficiency of an element of a screw propeller blade considered as an oblique plane moving transversely past the stern of the vessel. The discussion of the resulting equations brought out the principle that when the slip angle was proportioned so as to give best results, the best general efficiency was to be obtained when the blade was at an angle of 45° with the longitudinal. These results were reached by certain approximations, and by an assump-

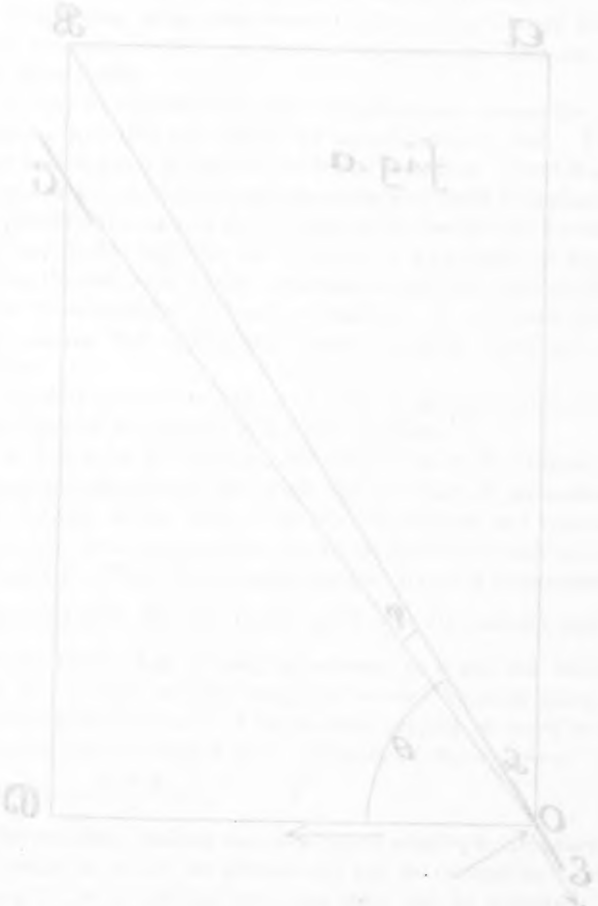
tion relative to the action of the friction, which was afterward recognized by Mr. Froude as incorrect or doubtful. The total quantitative influence of these departures from exactness was readily seen to be very small, however, and therefore the general correctness of the result is unquestioned. The double conditions required for the attainment of the highest efficiency, however, are such as can be fulfilled for one point only of a propeller blade, or, rather, by points having but one particular radius. In other words, due to the fact that the points on a propeller blade have varying transverse velocities as we go from the hub outward, while their longitudinal velocities are all the same, the proportions necessary for the highest efficiency are attainable only at one ring of points on the blades. For points within and without this particular radius the efficiency will be less, and therefore for any blade as a whole the efficiency will be less than that possible for a single element.

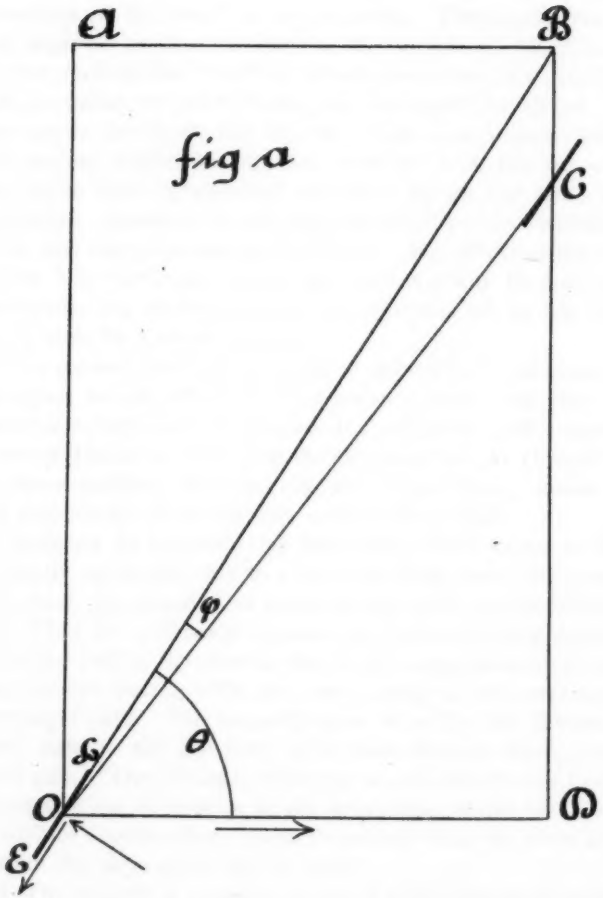
It has seemed, therefore, a matter of interest to re-calculate the expression for the efficiency of a plane element, and then by numerical substitution to compute the values for such a variety of circumstances as shall give the efficiency for any element of any screw working with any slip, the ranges being limited to such proportions as are actually common in practice.

It must not be forgotten that the results which we are to find are strictly applicable only to a screw working under the conditions which are assumed as fundamental, which are as follows:

1st. That the resistance opposed to a plane element moving in a helical path is the same as that found experimentally for the same element moving with the same velocity at the same angle in a straight path. This necessity arises from the lack of experimental data on the resistance of a plane element moving in a helical path. The principal difference is undoubtedly due to the difference in the disposition of the stream lines in the two cases, and various considerations seem to indicate that the error arising from this supposition will be small.

2d. The element is supposed to act in water previously undisturbed. That is, the disturbance due to the previous passage of the ship, and the mutual interference of one blade with another,


A
B
C
D
O
1
2
3
θ


A
B
fig a
C
φ
θ
O
D
E

are both neglected. See on this point a paper by Cotterill, in "Transactions of Institute of Naval Architects," Vol. XIX, or "Navy Scientific Papers," p. 126.

3d. The element is considered as without thickness.

It is hoped, in the near future, to carry on at Cornell University researches which may furnish experimental data on some of these points.

It may be objected that such simplifications remove the problem far from that with which we have actually to deal. This is not denied, but it is urged in reply that the whole subject of screw propulsion is so indefinite and unsatisfactory, both in fundamental experimental data and theory, that no excuse should be needed for any further light on the behavior of a propeller or any element thereof, even under conditions which are confessedly far from those actually obtained in practice. It is by such gradual approaches that indications toward valuable truth are often found.

We will proceed, therefore, to deduce the expression for the efficiency of an element of a screw propeller.

In Fig. *a* let EL represent the edge view of the element, OA being the direction of the shaft and OD that of the motion of EL relative to the ship. Let the longitudinal and transverse velocities be so related that the actual developed path of EL is along OC. Then if OD is the distance traveled transversely in one revolution, BD will be the pitch $= p$, BC the slip, and $\frac{BC}{BD}$ the slip ratio. Let the angular velocity be ω and the radius of EL be r. Then velocity along $OD = r\,\omega$. Velocity along $DC = r\,\omega \tan(\theta - \varphi) = u$. Longitudinal velocity of screw or element $= BD = r\,\omega \tan\theta = v$. Velocity of slip $= v - u$. Slip per cent. $= \frac{v-u}{v} = s$.

Now without making any assumption relative to the exact law according to which the element EL will be resisted as it moves along OC, it is sure, at least, that there will be resistance, and whatever the size and direction of the total resistance of such motion, we can replace it by two resistances, one normal and

one tangential, the latter being due to fluid friction. Let P be the normal component and Q the tangential. The total longitudinal component of these must equal the thrust; or

$$T = P\cos\theta - Q\sin\theta, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The effort F, which must be exerted to produce this, must equal the total transverse component of P and Q, or

$$F = P\sin\theta + Q\cos\theta, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The useful work will be Tu and the total work $Fr\omega$. Therefore the efficiency is

$$e = \frac{Tu}{Fr\omega} = \frac{(P\cos\theta - Q\sin\theta)}{(P\sin\theta + Q\sin\theta)}\tan(\theta - \varphi), \quad . \quad . \quad (3)$$

The existence of Q is evidently prejudicial to efficiency, and therefore the highest value possible will be that corresponding to $Q = O$. We have then

$$e = \cot\theta\tan(\theta - \varphi) = \frac{OD}{BD}, \ \frac{DC}{OD} = \frac{DC}{BD}$$

or $e = 1 - s$.

From the entire generality of the preceding, it would seem to follow that the highest efficiency of which any propeller or element is capable is $(1 - s)$, rather than $\frac{1 - s}{1 - \frac{s}{2}}$, according to Rankine's original memoir, published in the "Engineer" of Jan. 11, 1867, and republished in the collection of his miscellaneous papers. The cause of this difference lies in the matter of shock, a subject whose relation to propeller efficiency the present writer hopes to discuss in a future article.

Returning to (3) we have next to assume laws controlling P and Q. The former is assumed of the form $P = bAv_1^2\sin\varphi$, b is taken at 1.7, and v_1 is the velocity along OC.

The expression for the frictional resistance on a plane moving in its own direction is taken of the form $F_1 = fAv_1^2$, where f is a constant called the coefficient of liquid friction, and v_1 is the relative velocity between plane and water.

In the present case, however, the element EL moves at an angle φ with its own direction. Lord Raleigh has given an expres-

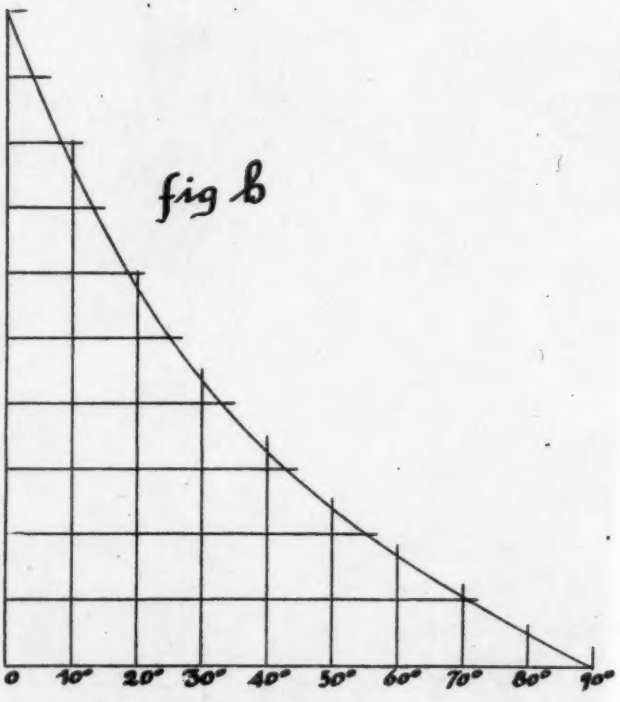


fig b
0
10°
20°
30°
40°
50°
60°
70°
80°
90°

sion for the point on EL at which the two streams will separate, one flowing around L, the other around E. The frictional resistance will be, then, the difference between the effects due to these two streams. The general result is to decrease the net coefficient of friction for the front surface of a blade moving obliquely through the water. The expression for the reduced coefficient f^1 in terms of f may be found in the "Transactions of the Institute of Naval Architects," Vol. XX, p. 152, or "Navy Scientific Papers," p. 134, and is as follows:

$$f^1 = f \frac{4 \cos \varphi - (\pi - 2\varphi) \sin \varphi}{4 + \pi \sin \varphi}.$$

The computation of the coefficient expressed by the fraction gives a series of values which are plotted in Fig. *b*. It may be seen that the value of f_1 rapidly decreases as the blade becomes slightly oblique. For the back side of the blade the friction is undoubtedly still less, but as to how much, experimental data are lacking. The angle φ which measures the angles of obliquity is not constant, but varies for different values of the slip and radius. It varies but slightly, however, and in view of the general uncertainty in the exact value of the net coefficient of friction, the angle φ was taken as constant at a mean value. With feet, pounds and seconds as units, and .004 as the value of f for a single surface, the value .0068 was taken as the value of f^1 for both surfaces, front and back. We have, therefore, as values of P and Q the following:

$$P = 1.7\ Av_1^2 \sin \varphi.$$
$$Q = .0068\ Av_1^2.$$

Substituting in (3) we have:

$$e = \frac{1.7 \sin \varphi \cos \theta - .0068 \sin \theta}{1.7 \sin \varphi \sin \theta + .0068 \cos \theta} \tan (\theta - \varphi), \quad . \quad (4)$$

$$\text{Put } k = \frac{.0068}{1.7} = .004,$$
$$l = DC = (1 - s)p,$$
$$a = OB,$$
$$b = OC.$$

We may then readily reduce (4) to the following form:

$$e = \frac{4\pi^2 r^2 l^2 (1 - s)s - kl^2 ab(1 - s)}{4\pi^2 r^2 l^2 s + 4k\pi^2 r^2 ab(1 - s)^2}, \quad . \quad . \quad . \quad (5)$$

For numerical computation the following form is convenient:

Put $r = nl$,

$$a_1 = \sqrt{4\pi^2 n^2 + \frac{1}{(1 - s)^2}} = \frac{a}{l},$$

$$b_1 = \sqrt{4\pi^2 n^2 + 1} = \frac{b}{l}.$$

We then readily find

$$e = \frac{4\pi^2 n^2 (1 - s)s - k(1 - s)a_1 b_1}{4\pi^2 n^2 s + 4k\pi^2 n^2 (1 - s)^2 a_1 b_1}.$$

n and s were now treated as independent variables, the first fixing the location of the element, the second its slip. There were then computed for this expression sixty values covering ranges in n from .05 to .60, and in s from .05 to .30, both inclusive. The lower value of n represents a location well within the hub of ordinary propellers, and the outer one beyond their tips, so that we have in these sixty values the necessary data to give the efficiency of a point on any ordinary propeller with any ordinary slip. The value l or DC was taken as one of the variables instead of DB the pitch, because it is the distance which is essentially constant for a constant value of the angular velocity. The pitch may vary at every point on the blade, but the longitudinal distance traveled in one revolution is the same for all.

The most complete method of representing graphically the variation in e is by a surface whose ordinate above a certain base plane at a point determined by n and s as rectangular co-ordinates, shall be equal to the corresponding value of the efficiency. This was done for the whole range in s and for the range .05 to .35 inclusive for n. These ranges include all the characteristic variations in e, and give values for all points of all propellers except the tips of such as have a small pitch in proportion to their diameter. The method of procedure was as follows: A 40 per cent. level was taken as a datum plane, as nearly all the values of e came above this level. Knitting needles were then

cut and inserted in a block of wood ruled in squares, and adjusted so that their length above the block was equal to the value of e at that point. The scales were 1″ for a difference in n and s of .05 and 1″ for 10 per cent. in e. Modeling clay was then filled in to the tops of the needles and a fair surface smoothed off. Of this a plaster cast was taken in the usual way, which was used as a pattern for a brass casting, which finally was corrected and finished on the surface. On this were then ruled lines representing the intersections with it of the series of vertical planes at right angles to each other, each of such curves giving a curve between e and n for constant s, or between e and s for constant n. Level or contour lines were then ruled, giving the locations of the various conditions corresponding to constant e. Finally lines were ruled giving the locations of the various angles θ by 5° differences from 25° to 75°.

No graphical representation in two dimensions will quite take the place of such a three-dimensional model, but the variation of e may be studied with relative ease by means of the various sections of this surface by the three series of planes. These are given in Figs. 1, 2, 3, 4.

Fig. 4 showing the contour lines gives the best idea of the form of the surface and therefore of the nature of the variation in e due to any proposed change in either the location or slip of the element. The general nature of the remaining portion of the surface—that for the tips of the blades whose pitch is only slightly greater than their diameter—is easily seen from an examination of the contour lines near the outer end in Fig. 4 taken in connection with the section curves for the .40, .50 and .60 values of n given in Fig. 2. With the exception of these three, all of the curves in the various figures refer to the portion within the value $n = .35$.

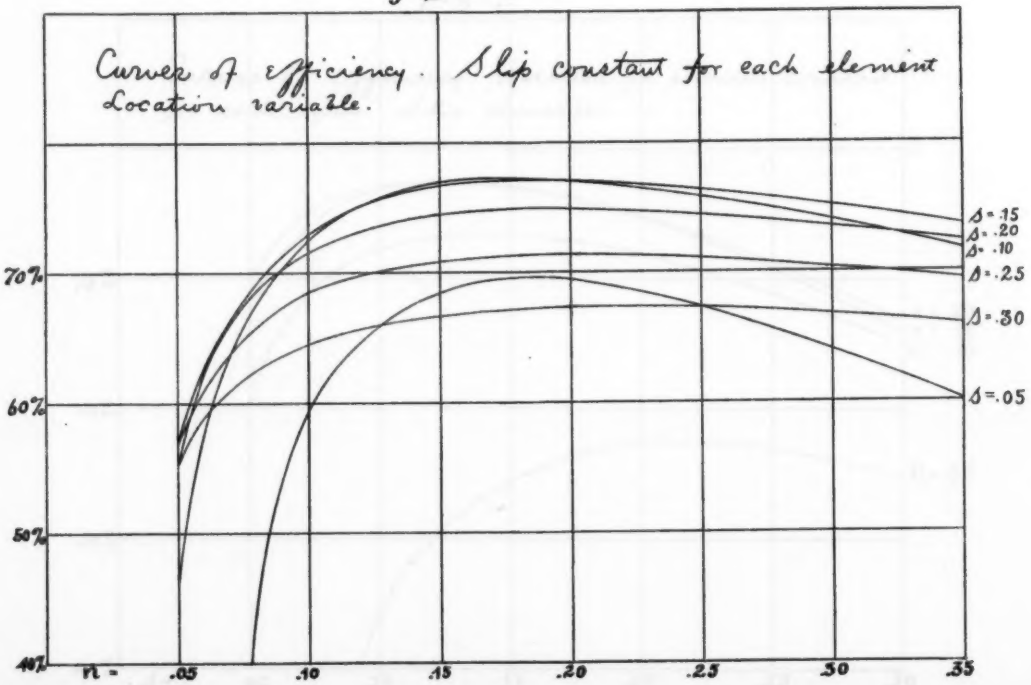
The diagrams need but little explanation. It will be noticed how gradually e falls off on passing beyond either the location or slip for the best results, while within such values it decreases with great rapidity.

The dotted line ab of Fig. 4 marks the location of the crests of the curves of figs. 2 and 3. That is, it gives the best slip for

each location of the element. It would appear from this that the slip should first decrease to a minimum of about $12\frac{1}{2}$ per cent. at the location of maximum efficiency, and then slowly increase again towards the tips of the blades. The pitch which equals $\frac{l}{1-s}$ would therefore be maximum near the hub, then decrease rapidly to a minimum near the 45° location, where the slip is also minimum and the efficiency maximum, and then slowly increase from this point on to the tips of the blades. As far as the writer is aware, this law for radial variation of pitch is different from any yet proposed, and while it is possible that the variations in the case as here assumed for mathematical discussion from an actual case would be such as to obscure any benefit to derived from such an arrangement, yet there seems to be as be much reason for such a law of pitch variation as for any of the other various laws proposed on this point.

Figs. 1, 2 and 3 show that, for small values of the slip or a location near the hub, the efficiency curves dip down very steeply. As a matter of fact they would meet the o level before either n or s become o. For values of n or s within these points we should have e a negative quantity. To see how this can be we turn to the value of e and note that for $e = o$ or negative, the condition is $P \cos \theta$ equal to or less than $Q \sin \theta$. Reducing the condition for $e = o$ we find $4\pi^2 n^2 s = ka_1 b_1$, which may be solved for either n or s. The actual solution presents no especial interest, since such points, in ordinary cases, are either well within the hub or else correspond to a smaller slip than ever met with in practice. To arrive at the physical meaning of a negative value of e, we note that as n is decreased, s remaining constant, the direction of the element becomes more and more nearly fore and aft, and the longitudinal component of the reaction experienced will rapidly decrease, while proportionately the same component of the frictional resistance will increase. At some point the two will become equal, and within this point there will be an actual longitudinal drag, the thrust becoming negative. Such an element would not, of course, be self-propelling, but we may readily imagine such an element as part of

fig. 1


Curves of efficiency. Slip constant for each element
Location variable.
70%
60%
50%
40%
n =
.05
.10
.15
.20
.25
.30
.35
δ = .15
δ = .20
δ = .10
δ = .25
δ = .30
δ = .05

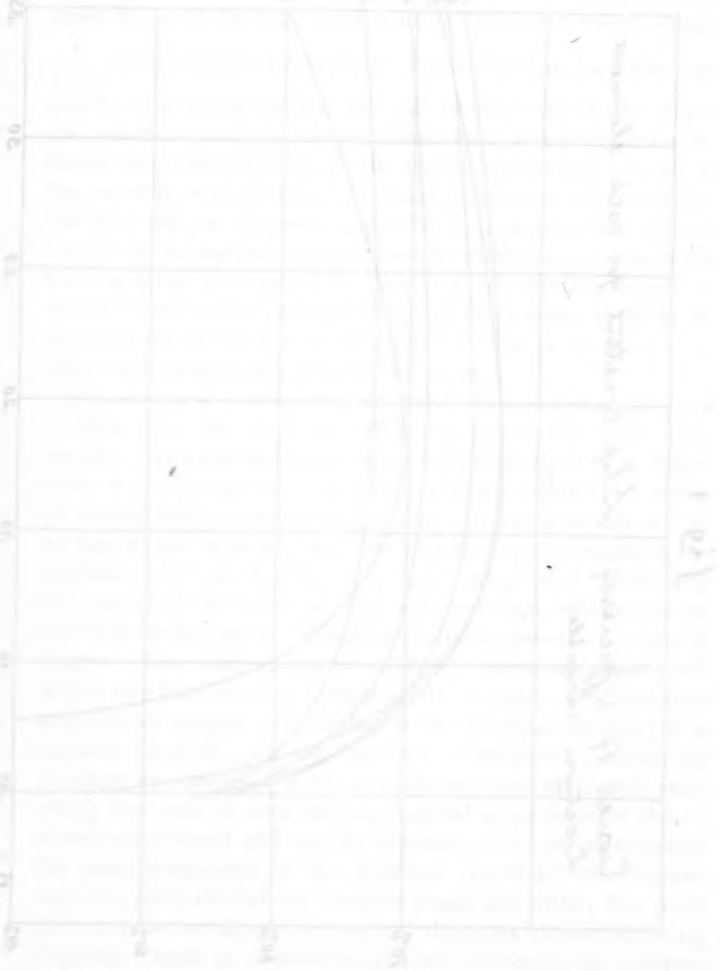
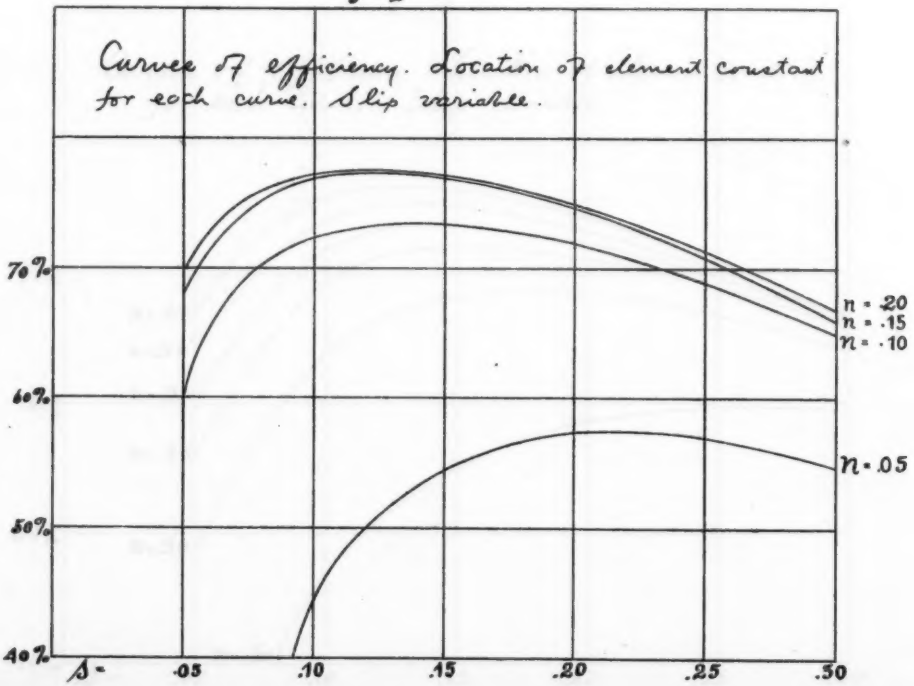


fig. 2


Curves of efficiency. Location of element constant for each curve. Slip variable.
70%
60%
50%
40%
s = .05 .10 .15 .20 .25 .50
n = .20
n = .15
n = .10
n = .05

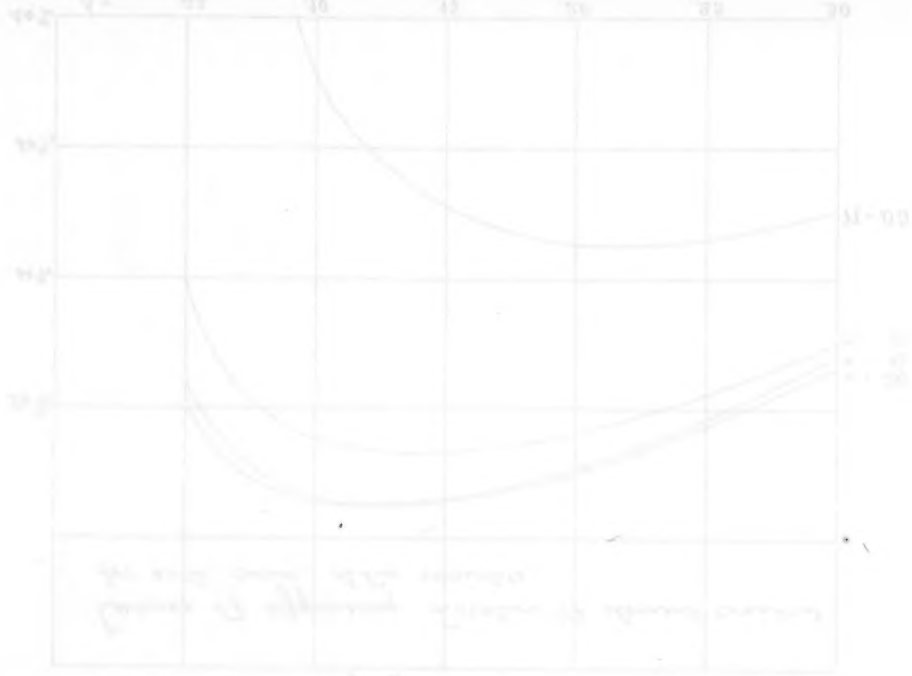
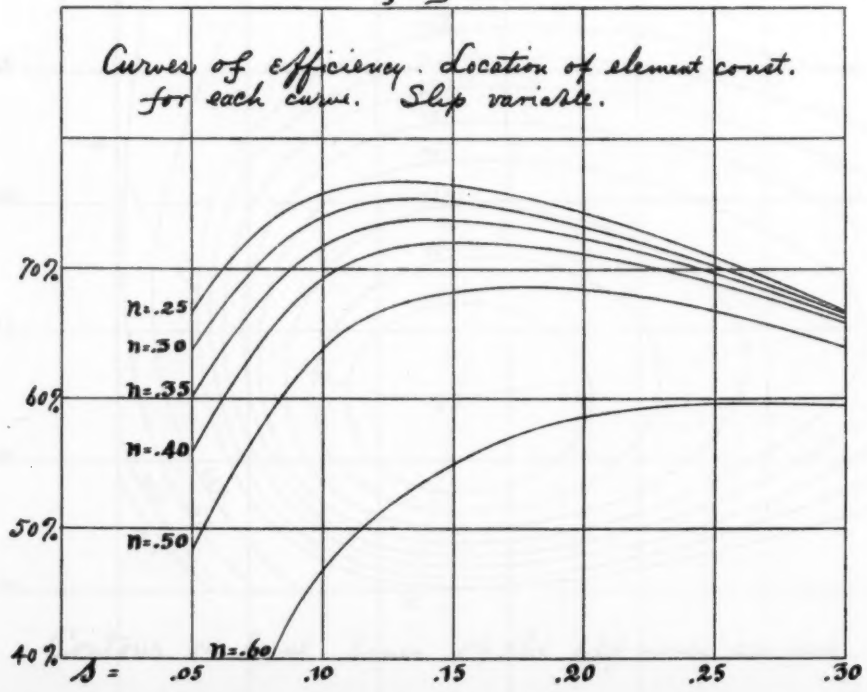


fig. 3


Curves of efficiency. Location of element const. for each curve. Slip variable.
70%
60%
50%
40%
n=.25
n=.30
n=.35
n=.40
n=.50
n=.60
S = .05
.10
.15
.20
.25
.30

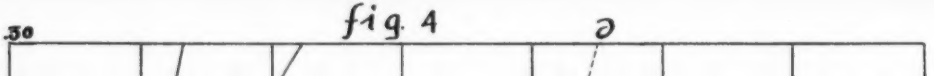
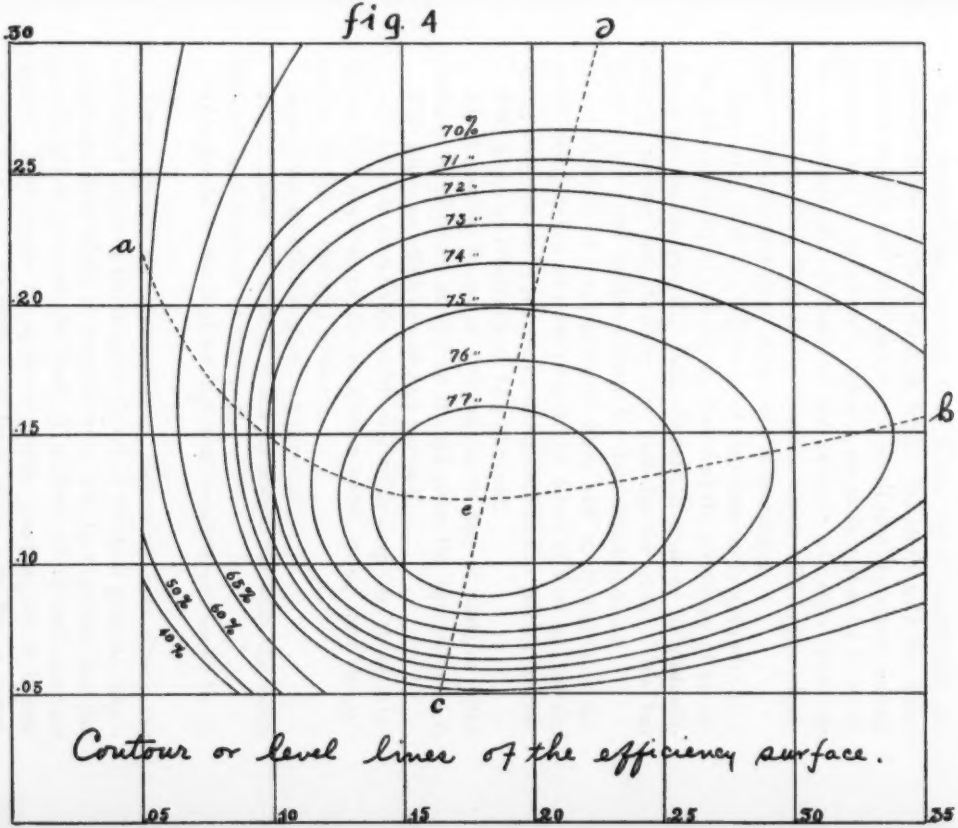
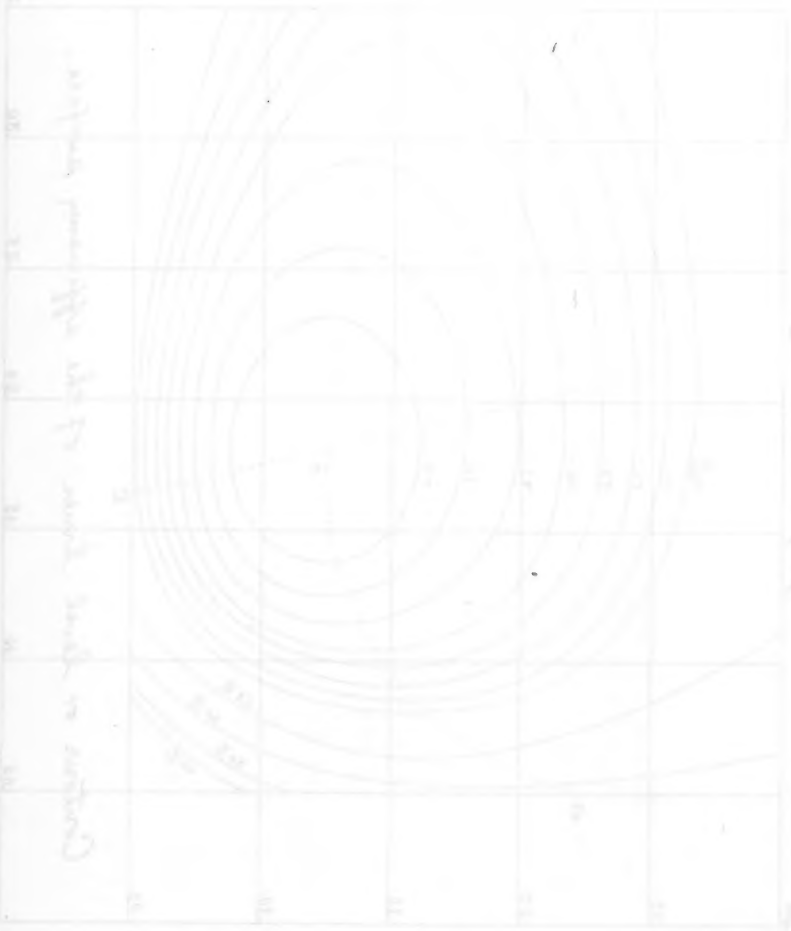


fig. 4

fig. 4



Contour or level lines of the efficiency surface.



a screw and dragged through the water at the expense of the reaction gained by other elements more advantageously situated.

In a similar way, we see that if the slip is indefinitely decreased, there will be a point at which the angle with the direction of motion will have become so small that the actual normal reaction will be very small. Hence, no matter what its location, the longitudinal component of the reaction will equal and then become less than that due to friction, which will, in the meantime, increase somewhat as the slip is decreased.

Assuming that the efficiency of one element is not influenced by the contiguity of others, we may find the efficiency of an entire blade by multiplying each element of area by its own efficiency, adding and dividing the sum by the area. This has been worked out approximately for the following cases:

1st. A blade of uniform width from hub to tip, the inner element being taken at $n = .10$ and the outer at $n = .45$, the slip being constant at 15 per cent., and therefore the pitch constant.

This gives an efficiency of 74.3 per cent.

2d. A similar blade of uniform width but of variable pitch radially, so disposed as to give at each point the best value of e.

This gives an efficiency of 74.5 per cent.

3d. A blade of variable width of similar shape to those fitted to the new naval ships, the maximum width being near the 45° angle, the pitch and slip constant.

This gives an efficiency of 74.9 per cent.

4th. A blade of variable width as above, but with the pitch variable and disposed as in No. 2.

This would be the best arrangement possible, and we find $e = 75.1$.

These results are instructive as indicating how slowly the efficiency varies in the neighborhood of the best general disposition, and how slight an improvement is to be expected over the results given by a simple blade of uniform width and constant pitch. This seems to agree well with practice, for it is well known how wide a variation in form, pitch, slip, &c., is sometimes admissible with but slight change in efficiency.

It will be noted that in this discussion no mention has been

made of axially expanding pitch. The paper being essentially a study of an element, such an examination comes hardly within its scope; it may be remarked, however, that before such forms can be treated by the method used in the present paper, further experimental data on the passage of curved blades through the water are needed. The reason usually assigned for such pitch variation, viz., the avoidance of shock, involves a question already mentioned in the early part of the present paper.

Another point which may be noted as flowing directly from the form of the expression for e has reference to the distribution of I. H. P. among its various portions. It is readily seen that the part expended on screw friction must be eliminated before the power spent on the screw can be divided into two parts proportional to s and $(1 - s)$, the former of which gives the work spent on the slip, the latter that on the ship.

The study of an element of a screw propeller cannot, of course, take the place of a study of the screw itself. It is believed, however, that further experimental data on the passage in a helical path of elements having thickness, and with plane and curved faces, would furnish the foundation for a fairly complete theory of a screw propeller element, and that such a complete theory could not fail to give indications toward valuable truths relative to the propeller itself.

IV.

THE FAILURE OF STEEL CASTINGS FOR THE U. S. S. *MAINE* AND *CRUISER No. 11.*

By Assistant Engineer C. A. Carr, U. S. Navy.

The following article is intended to give the history of the success or failure of the steel castings furnished to the Quintard Iron Works for the U. S. S. *Maine* and *Cruiser No. 11.* These castings had previously been inspected at the works at which they were made, and were, to outward appearances, perfect. The faults which led to their rejection developed either while being machined or when the parts were subjected to the hydrostatic tests required by the specifications.

It is hoped that it will call out an article from an inspector who has been for some time at works where such castings are made, from which we shall be better able to judge of the advance which has been made in casting steel within the past two years, and, perhaps, learn something of the changes in the process of manufacture which have brought about this advance.

It will be seen that the clauses in the machinery specifications of naval vessels now building, requiring certain parts to be made of cast steel, have cost a good deal, both in money and time, and often in the end the Department has been compelled to give up cast steel and use some other material. In the case of the Y-frames for the *Detroit* and the *Montgomery*, its use has led to an increase rather than a reduction of weight. Still, reduction of weight in marine machinery being a subject of such vital importance, the cost of the encouragement given to the development of this industry will, doubtless, in the end, well repay the concessions made.

For much of the data and description to be given, I am indebted to Chief Engineers Morley and Heaton and Assistant Engineer Moritz, of the U. S. Navy, and to Mr. N. F. Palmer, Jr., of the Quintard Iron Works.

The first column of the following table gives the principal parts which were originally required to be of cast steel or in which cast steel was optional, and the second column the material used in those parts as completed:

U. S. S. MAINE.

PARTS TO BE OF CAST STEEL AS ORIGINALLY DESIGNED.	MATERIAL OF SAME PARTS AS COMPLETED.
Main engine columns	All of wrought steel.
Inverted Y-frames I. P. cylinders	Both of cast steel.
Main engine bed plates	All of cast steel.
Valve-chest covers	18 cast iron, 6 cast steel.
Cylinder and valve-chest linings	All of close-grained cast iron.
Valve stem guides, H. P. cylinder	Cast steel.
Equalizing bars, valve gear	Cast steel.
Rock-shaft arms, valve gear	All of wrought steel.
Rock-shaft brackets, valve gear	Cast steel.
Valve stem cross-heads	Cast steel.
Reversing-shaft arms	Cast steel.
Reversing-shaft brackets	Cast steel.
Separators, fire room	Cast steel and wrought steel, as specified.
Separators, engine room	Cast steel, wrought steel and composition.
Turning wheels	Cast steel.
Gear wheels for windlass	Cast steel.
Racks at bottoms of turrets	Cast steel.
Stiffening rings upper man-hole plates	Cast steel.
Man-hole and hand-hole plates	Cast steel.
Pipes through double bottom	Partly cast steel, partly composition.

Main Engine Columns.—Each high and each low-pressure cylinder was intended to be supported on four hollow cast-steel columns, and each intermediate-pressure cylinder on two cast-steel columns and an inverted Y-frame. These columns were about 7 feet 4 inches long, and 7 inches outside diameter, the metal to be 1 inch thick, and the ends to terminate in collars $1\frac{3}{4}$ inches thick for securing to the cylinders and bed plates. These columns had facings for brackets for carrying the lower end of cross-head guides, and bosses to receive fore and aft and athwartship tie rods. Twenty-five columns were received at the works. Five were condemned on account of blow and sand holes, being

cracked at the flanges, or the flanges being reduced too much or broken in cutting runners. The remaining twenty, a complete set, were at first accepted and erected in place in the shop. After being erected, it was found that two of them showed evidences of having been hammered about the bosses to close seams. This led to taking them down and machining them. The first cut developed seams about the bosses, also blow and sand holes; and the columns were taken down one by one and machined with similar results, from which the following samples are selected:

No. 2.—Bad seam in upper boss. Sand and blow holes the whole length of column.

No. 6.—Seam in the upper boss. Small seam at the lower flange. Blow and sand holes.

No. 11.—Seam in upper boss, extending all the way through. Large blow and sand holes between bosses.

No. 14.—Seam $4\frac{1}{2}$ inches long on the lower side, one $1\frac{3}{4}$ inches long on upper side of lower boss, and one on the lower side of the upper boss extending almost around. Blow and sand holes on bosses.

No. 15.—Seams at bosses, the one at the middle boss extending half way around. Seam at the lower flange. Blow and sand holes.

No. 17.—Seam in the fillet of the upper flange. Seam between the lower flange and the boss. Blow and sand holes.

No casting was good, so that, out of the twenty columns, only two could be accepted, and the character of these two castings was such that it was decided to reject the whole lot and substitute instead wrought-steel columns. The time from the giving of the order for the cast-steel columns to the rejection of the last one was fourteen months.

Inverted Y*-Frames for I. P. Cylinders.*—These frames are of simple form, the cross-head guide being supported by the columns, and the cross-section at any point of the length an I. Both castings were good.

Main Engine Bed Plates.—These consist of three steel castings, flanged, bolted together, and faced for the columns and

main bearings. None of those received at the shop were rejected. Extra work was required on them, however, on account of lack of knowledge as to the allowance to make for shrinkage.

Cylinders and Valve Chest Linings.—Cast steel was optional in this case. Close grained cast iron was chosen, because the firm knew, from previous experience on the *Concord* and the *Bennington*, the loss of time there would be in obtaining the necessary castings perfect from cast steel.

Valve Chest Covers.—These were to be of cast steel of dished form. Thirty-four of these castings were received at the shop, and of this number thirteen were rejected on account of seams and blow and sand holes which developed while being machined. Fifteen more were rejected after being tested under the required water pressure. The metal was very spongy about the center and thickest part of the covers. In several cases, what appeared to be merely surface defects proved to be blow holes that extended entirely through the metal. The last cover was rejected thirteen months after the order was given to the steel maker. It was then decided to use cast iron in place of the cast steel ones found defective.

The *Rock-Shaft Arms* for the valve gear delivered at the shop were rejected on account of seams and sand and blow holes, and wrought steel ones substituted.

Engine Room Separators.—The flange ring is the only part of cast steel, the top being of composition.

Turning Wheels.—One worm wheel was rejected on account of being full of sand and blow holes.

Gear Wheels for Windlass.—These are worm wheels about 4 feet 8 inches in diameter. Two were rejected on account of blow holes.

Man-hole Plates for Boilers.—Four were rejected when tested under water pressure. Their state was similar to that of the valve chest covers described above, the spongy places and blow holes being where the metal was increased in thickness to receive the handles.

Pipes through Double Bottom.—It was found very difficult to get castings free from cracks and serious defects on account of

sand and blow holes. Eight castings were rejected, and now the smaller castings are mostly of composition and the larger ones of cast steel.

Other castings than those specially mentioned here were rejected, but they were replaced without any considerable delay.

Considerable difficulty has been found at the Quintard Works in making proper allowance for the shrinkage of steel castings, even when the shrinkage for the same or similar patterns was known. In some cases an allowance of $\frac{1}{4}$ inch to the foot has not been found too much. The allowance for shrinkage in the columns of the *Maine* was $\frac{3}{16}$ inch to the foot. The actual shrinkage in length was almost nothing; and in order to use the castings it was necessary to raise the cylinders three-quarters of an inch and increase the length of the connecting rods and eccentric rods by the same amount. This particular case may be explained by the shape of the casting and the character of the mold.

About 36 tons of rough steel castings have been used at these works for the machinery of the *Maine*. To supply this amount, about 52 tons have been received; that is, 44 per cent. of the total amount required was rejected, after a great deal of machine work had been done.

CRUISER No. 11.

PARTS TO BE OF CAST STEEL AS ORIGINALLY DESIGNED.	MATERIAL OF SAME PARTS AS COMPLETED.
Main engine frames	Built-up plate steel.
Main engine bed plates	Cast steel.
Valve-chest covers	Cast steel.
Pistons	Cast steel.
Cross-heads	Built-up wrought steel and composition.
Valve-stem cross-heads	Cast steel.
Valve stem guides	Cast steel.
Eccentrics	Cast steel.
Reversing-shaft arms	All but three of cast steel.
Reversing-shaft brackets	Cast steel.
Stern tube-supporting rings	Cast iron.
Air pump engine frames	Wrought steel.
Turning wheels	Cast steel.
Boiler man-hole stiffening rings	Cast steel.
Boiler man-hole plates	Cast steel.
Separators	Cast steel, wrought steel and composition.

Engine Frames.—Each cylinder was to be supported on two inverted Y-frames of cast steel similar to those shown in the plate at the end of the article for *Cruisers Nos. 9* and *10*, except that they were to be in one piece instead of three, and were to be made much lighter. The contract for making the frames as originally designed was in the hands of the steel casting company for six months. After making many trials they acknowledged that they had been unable to produce a single sound casting as required, and asked for authority to cast them in three pieces. This plan was adopted for *Cruisers Nos. 9* and *10*, but for *Cruiser No. 11*, built-up plate steel frames were substituted.

Main Engine Bed Plates.—These are made in three castings, flanged and bolted together, with facings for crank-shaft brasses and caps and for the flanges of the engine frames. One section was rejected on account of cracks on each side of crank-shaft bearings.

Valve-Chest Covers.—All are of cast steel. None failed under the water pressure test, and all but two have been tested. Two covers and five caps for covers were rejected on account of sand and blow holes. The valve-chest covers for *No. 11* are flatter than those for the *Maine*.

Pistons.—Three I. P. Pistons were rejected on account of deep blow and sand holes near the center of the piston. One L.P. piston was rejected on account of blow holes at the center, and another on account of blow holes from 1 inch to $2\frac{1}{2}$ inches deep at the inner edges of the follower seat. One I.P. and three L.P. piston followers were rejected because they were full of sand and blow holes.

Cross-heads.—Four of the cross-heads were rejected because deep blow holes were found under the runners at the top of the wearing surfaces of the connecting-rod journals. Some of these blow holes were as much as $1\frac{3}{8}$ inches deep. The fifth was free from blow holes, but had a bad sand defect on the wearing surface of the connecting-rod journal. The sixth was never machined, it having been decided to substitute built up wrought steel and composition for the cast steel. No defects sufficient to warrant rejection were developed in finishing the shoe surfaces.

Valve-Stem Cross-heads.—Six valve-stem cross-heads were rejected on account of shrinkage cracks and sand and blow holes.

Eccentrics.—Seven halves of main eccentrics were rejected on account of deep blow holes.

Reversing-Shaft Arms.—Nine reversing-shaft arms were rejected because of deep sand and blow holes where keyed on the shaft. The last three defective castings were replaced by forged steel to save time.

Stern Tube-Supporting Rings.—Cast iron was substituted at the request of the contractors to save time.

Air-Pump Engine Frames.—Owing to the certainty of trouble and delay in getting the necessary steel castings, the contractors were allowed to substitute wrought-steel frames of their own design.

Separators.—The only cast steel parts in the separators are the flange rings at the top. The bottom part is of wrought steel and the top of composition. Two flange rings were rejected on account of being full of blow holes.

Two manhole-plate stiffening rings and one L.P. follower have not yet been received, although work on the ship has been in progress for over two years.

From the above data it is evident that the percentage of material rejected, including the engine frames, does not differ much from that given for the *Maine*. Also, that considerable work was done on castings afterwards rejected.

COST OF STEEL CASTINGS.

The cost of steel castings of course varies with their size and the difficulty of making them. From the data available, it is fair to say that, for rough castings for marine engine work, cast steel costs four times as much as cast iron. But this does not represent the whole difference of cost. In order to increase the probability of getting sound castings, and to allow for uncertainties in regard to shrinkage, the rough castings are made heavier than would seem necessary to one not acquainted with the material, and hence a greater amount of machine work is required to finish a steel casting. The average weight of the cast-steel

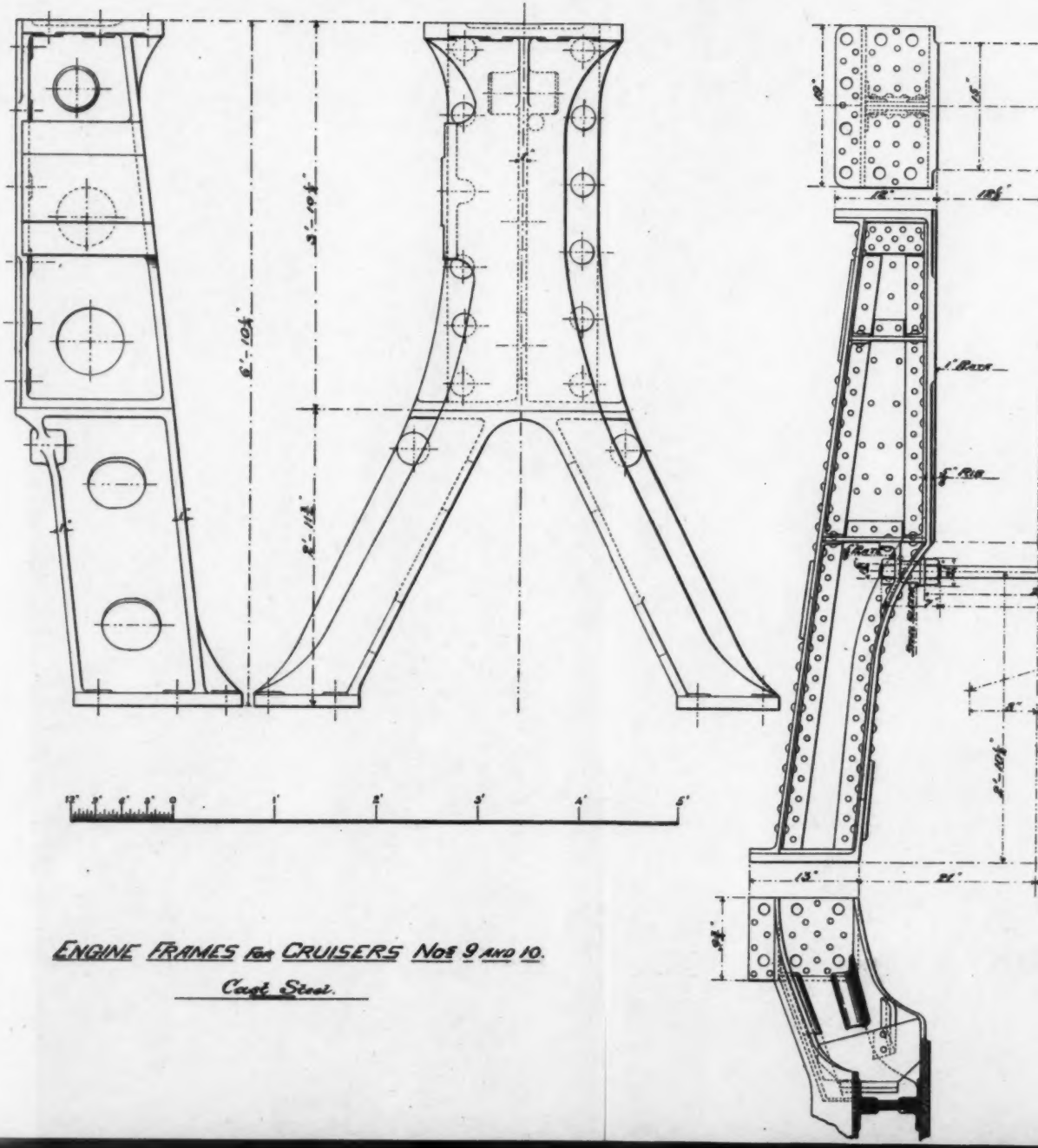
valve-chest bonnets for the *Maine*, as received from the foundry, was 362 pounds; the average finished weight was 302 pounds; that is, 20 per cent. of the finished weight was removed in finishing. About the same percentage of material was removed in machining the rejected cast-steel cross-heads for *Cruiser No. II*. Besides this, a further allowance for increased cost must be made to cover the uncertainty as to the soundness of the castings, the chance of doing a large amount of useless work on unsound castings, and the probable loss of time in contracts where there is a penalty for exceeding a certain time limit.

ENGINE FRAMES FOR *CRUISERS NOS.* 9, 10 AND 11.

These frames, as originally designed, were to be of cast steel, each frame being cast in one piece. They were to be 6 feet 5½ inches high, the flanges at the top and bottom 1½ inches thick, the vertical rib back of the wearing guide ¾ inch thick, and the metal elsewhere ⅝ inch thick. The calculated weight of one frame of this design without the cast-iron go-ahead wearing guide and tie rods, is 949 pounds.

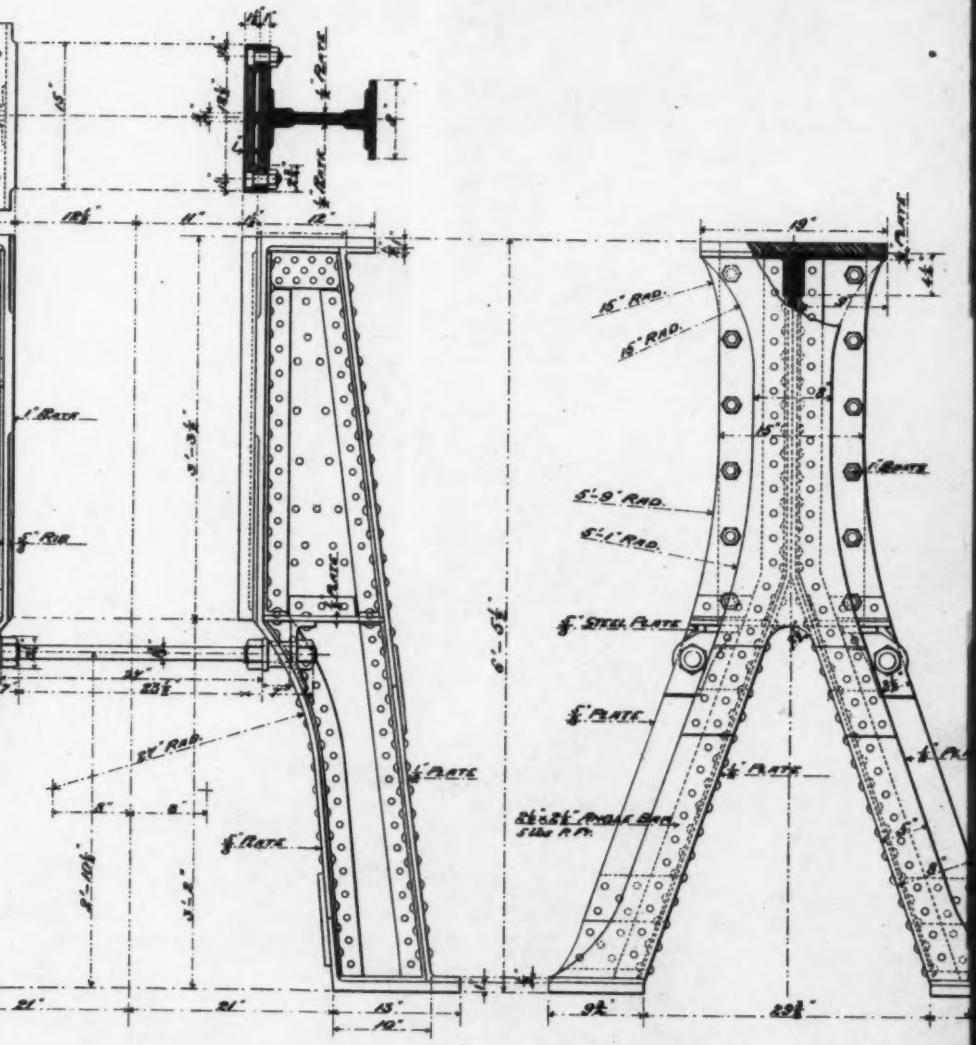
The engine frames shown in the plate as for *Cruisers Nos. 9* and *10* are of the design submitted by the steel casting company after their total failure to produce sound castings of the original design. They are cast in three pieces, as shown, and the thickness of metal is increased throughout. The actual weight of one frame without the go-ahead wearing guide and tie rods is 1429 ~~2,129~~ pounds. A similar frame of cast iron, with the same dimensions throughout and a 1-inch rib where the sections join, would weigh about 1,828 pounds. This gives a saving in weight of about a ton and a half on the engines of one ship in favor of cast iron, and they could have been built for about one-fifth the cost of the cast-steel frames.

The Quintard Iron Works preferred to use built up wrought-steel plate frames on *Cruiser No. II*, and this change was approved. These frames are models of good workmanship combined with lightness and strength, and are much admired by engineers who have seen them. They are shown in the plate, all principal dimensions being given. Cast-iron formers were



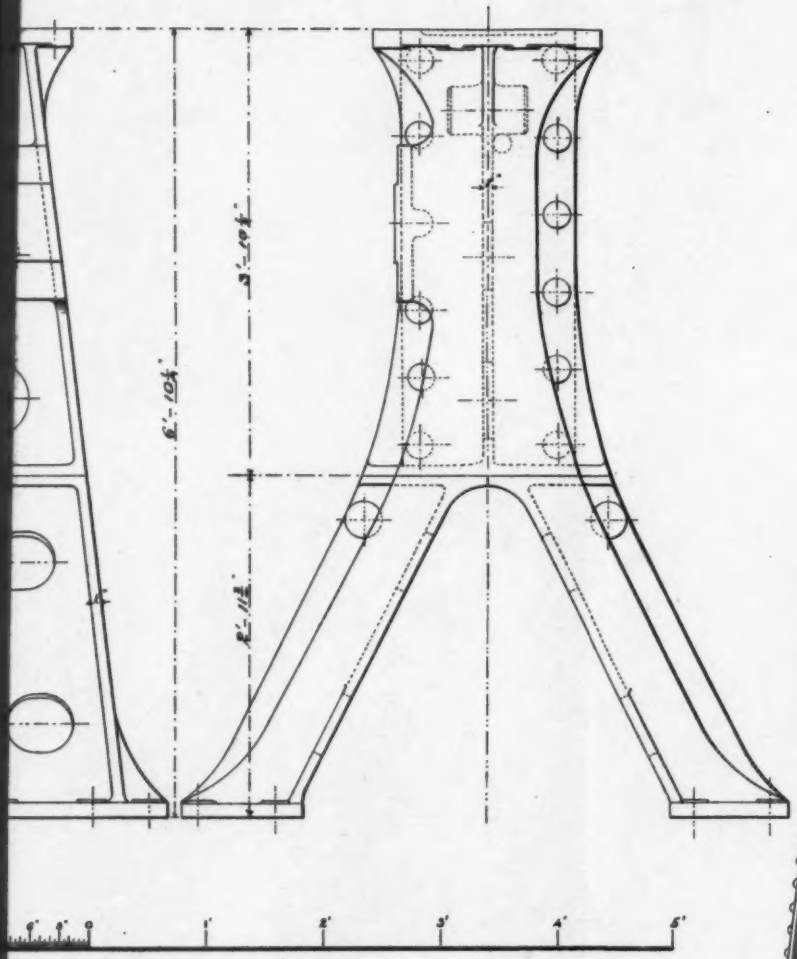
ENGINE FRAMES for CRUISERS Nos 9 AND 10.

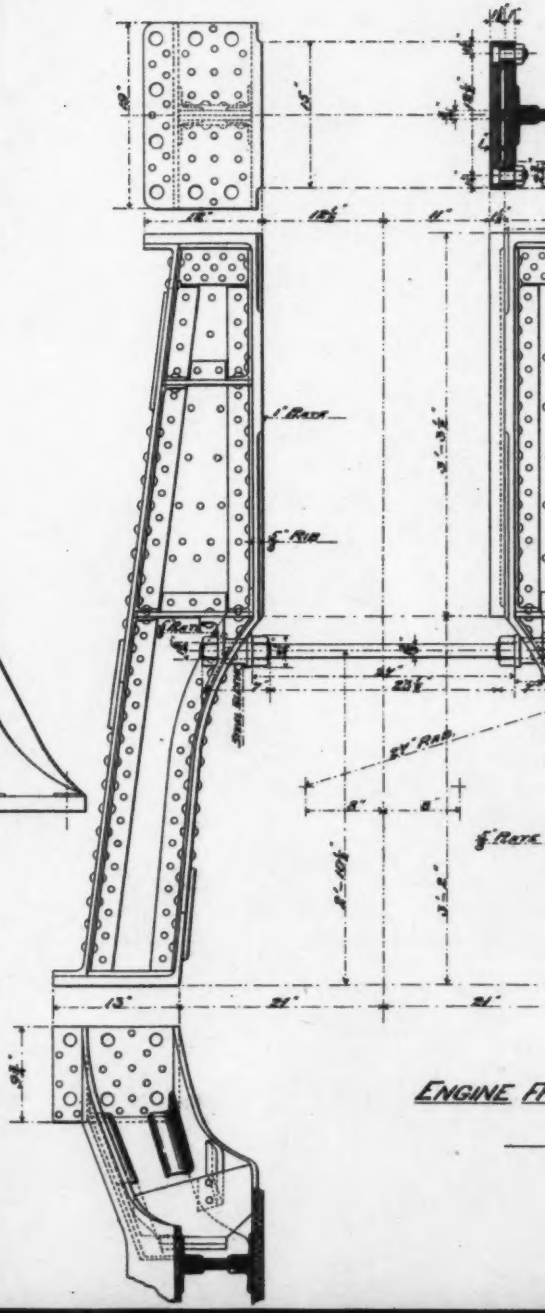
Cast Steel.



ENGINE FRAMES FOR CRUISER No: 11.

Wrought Steel.

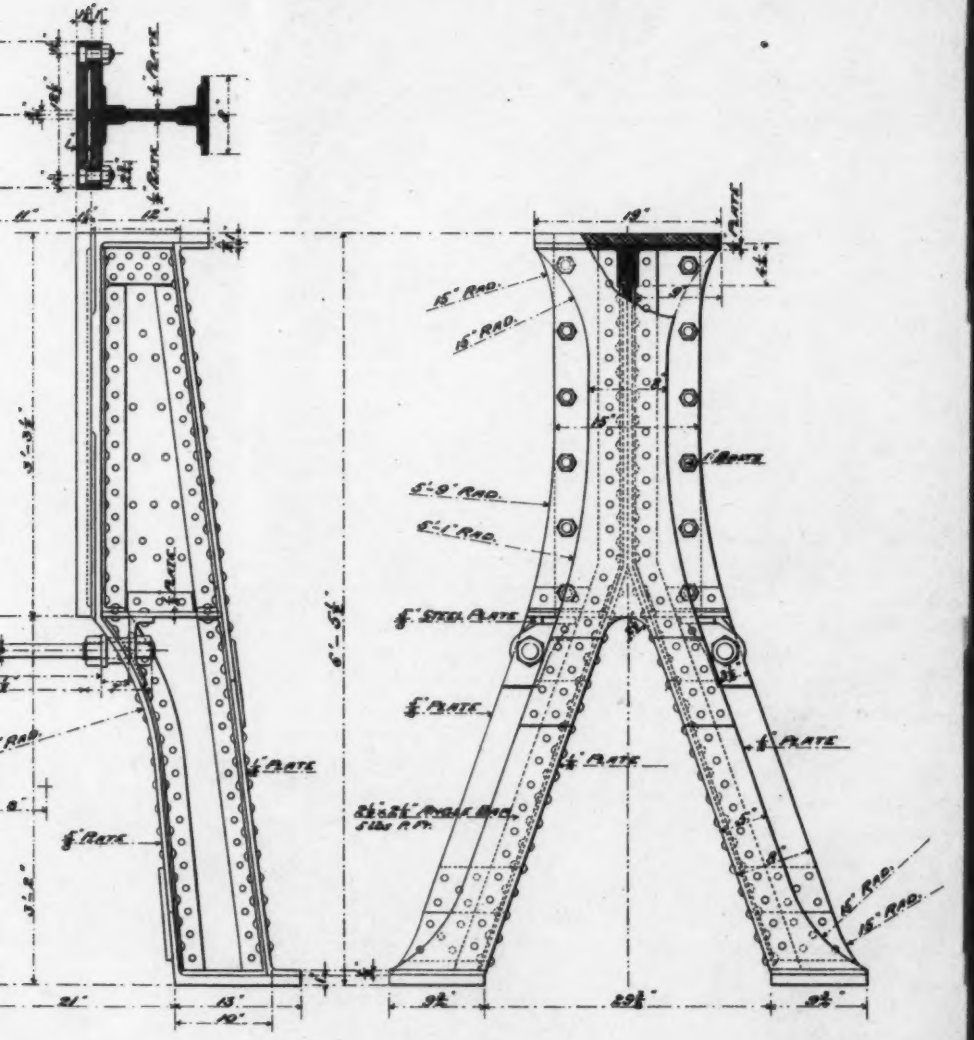




INE FRAMES FOR CRUISERS Nos 9 AND 10.

Cast Steel.

ENGINE F



ENGINE FRAMES for CRUISER No: 11.

Wrought Steel.

214

used to shape the outer and inner plates, and the angles were fitted in place on each frame separately. The frames were then erected and bolted together, and the rivet holes drilled; $\frac{5}{8}$ inch rivets were used throughout, being driven by hand. The rivets were driven hot and the heads formed by a button set.

It may be of interest to note here that the French Government have used built up frames on several of their gunboats, the rivets being made a driving fit and the riveting being done cold.

The calculated weight of one of the plate-steel frames for *Cruiser No. 11*, without the go-ahead wearing guide and tie rods, is 985 pounds, and the saving in weight of the two engines, as compared with the cast-steel frames for *Cruisers Nos. 9* and *10*, is about 6 tons.

These frames are symmetrical in appearance, and, on account of the symmetry, reduced weights, and less space required, afford greater accessibility to the machinery about them than would have been afforded by cast-steel or cast-iron frames.

They are undoubtedly much more expensive than cast-steel frames of the same weight, on account of the waste of material in cutting to the form, and the additional labor required to build them. As compared with the heavy cast-steel frames actually used, however, the difference of cost in favor of cast-steel frames is very much diminished, and is an item of small importance when we consider the great saving in weight by the use of built up plate frames, and the advantage of using a material which is reliable.

It may not be out of place to remark that the experience with steel castings for these two ships (which, moreover, is exactly the same for nearly all of our new vessels,) discloses a peculiar condition as to the art of steel casting. If it were an entirely new business, these repeated failures would not be so great a source of surprise and disappointment, but it must be remembered that steel castings of the same kind as most of those found defective have now been required for our naval vessels alone for over five years. It would certainly seem that this is a sufficiently long period to allow the determination of all the causes of failure and to discover remedies. Unfortunately, this does not appear to

be the case at all. The only reply of the steel makers is that they make great numbers of other castings without having any trouble, or, as in one case, that by making them of the same size as if of cast iron they can be made sound.

It is probably safe to say that only the imperative demand for the greatest possible reduction of weight in marine machinery has prevented the entire abandonment of steel castings.

I am sure that the pages of our JOURNAL will be freely offered to any of the gentlemen connected with the steel-casting works to explain these failures, and, as I trust, to encourage us to expect better and more reliable work in the future.

V.

BILGE DRAINAGE.

BY PASSED ASSISTANT ENGINEER WALTER F. WORTHINGTON, U. S. NAVY.

The arrangement of bilge drains may be divided into two general systems: 1st. Where the drain pipes are placed above the inner bottom of the ship; 2d. Where these pipes are placed below the inner bottom.

The first system appears preferable for several reasons. It permits the use of an open cistern, the condition of which can be inspected at any time. It permits the use of the well-known simple and efficient Macomb strainer. The valves are all necessarily more accessible for overhauling. The drain pipes are further removed from injury, in case the ship's bottom is damaged by rocks or torpedoes. It permits, owing to the open cistern, an easy solution of the problem of draining the flat floors, peculiar to iron and steel ships with inner bottoms, when

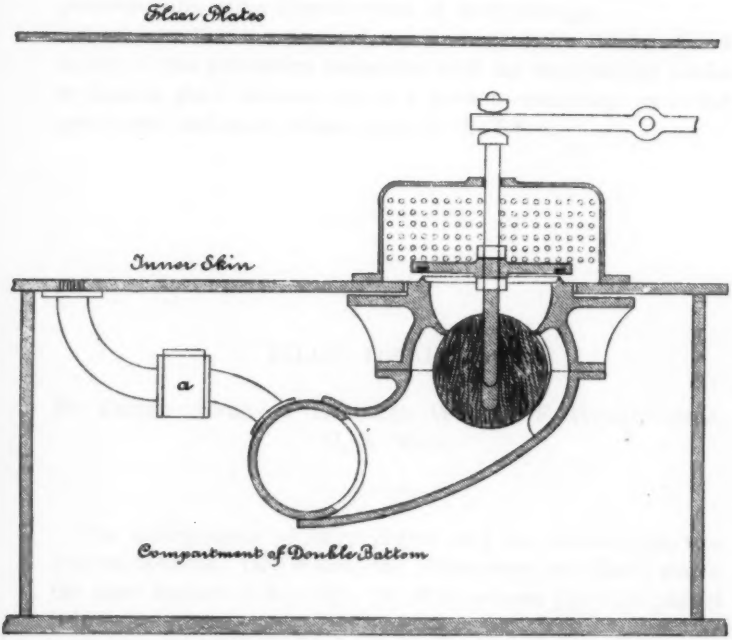


Fig 1

the ship is not on even keel. The compartment of the ship into which there is a leakage can be determined readily and certainly by examining the ends of the drain pipes which are perfectly accessible; with the other system water may back up from a distant compartment through the drain and leaky non-return valve, yet the fact could not be easily determined, owing to the wash of bilge water past and over the parts under examination. Finally, if any valve in the system leaked or was left open by mistake, the fact would be discovered by the increased difficulty of pumping the first time an attempt was made to pump the bilge; that is, the whole system would be submitted to a test at least once a day.

The scope of this paper is, however, confined to the second system.

To illustrate the subject, the system in use on two of our comparatively modern ships will first be described, its defects of design and workmanship pointed out, then improvements, suggested by several years' experience, proposed.

The general arrangement was as follows: A pair of parallel 12-inch drain pipes, extending as far fore and aft as the double bottoms, and passing through them, closed at the extreme forward and after ends by straight-way valves, and provided with straight-way valves to separate the parts under the engine room from the remainder.

In the forward fire room the connection beeween the drain and the inner skin was made as shown in the sketch, Fig. 1, by a cast-iron chamber containing a sphere of wood, intended to float up against a seat provided for it, and thus act as a non-return valve. The strainer was held down by three tap bolts passing entirely through the inner skin. Under the strainer was a disc of metal, provided with a rubber gasket, to serve as a valve, and actuated by levers, so that it could be closed from a level about ten feet above the fire-room floor, in case the non-return valve failed to work and allowed water to back up into the compartment. The strainers over the valves, starboard and port, were about three feet from the center line of the fire-room floor. A small pipe (about 2½ inches), leading to a fixed strainer

about half way between the other two, was provided with a butterfly valve in the chamber *a* to act as a non-return.

In the after fire room the arrangement was the same.

In the forward end of the engine room, in the double-bottom compartments, was placed a cistern, into which the large drains delivered, and which was divided fore and aft into two equal compartments which could be made to communicate by a sluice gate. The cistern was furnished with two manhole doors opening into the double-bottom compartments. A connection to the suction pipe of the steam and hand pumps, and to one main injection entered this cistern. From this cistern the drain pipes continued aft to another cistern at after end of engine room, in all respects like the first. The suctions of main and auxiliary pumps and after bilge injections entered it.

In the top of the after cistern, one on each side of the amidship partition and about 16 inches from it, were located the movable strainers as shown in the sketch, Fig. 2, and under these, enclosed in a cast-iron chamber, were the non-return valves of metal shaped like an inverted round pan. Over each strainer was placed a valve to close it by hand, in case the non-return failed to work. The yoke of this valve was so made that by slacking two nuts, adapted to be worked by hand, it could be readily swung out of the way and, if desired, altogether removed.

Returning now to examine the arrangement in the fire room, the objections are:

1. Only the lowest row of holes, not the broad bottom of the strainer, is available for drainage until the water accumulates, thus necessitating a larger strainer than would be required if it was inverted and placed with the bottom on a level with the inner skin.

2. The position favors the clogging of the holes, for as the lower holes stop up air continues to be drawn down through the top, lessening the force of suction which would otherwise draw small obstacles through.

3. The accumulation of water in the bilge, often the first indication we have of the fact that the strainer is choked, adds greatly to the difficulty of cleaning it.

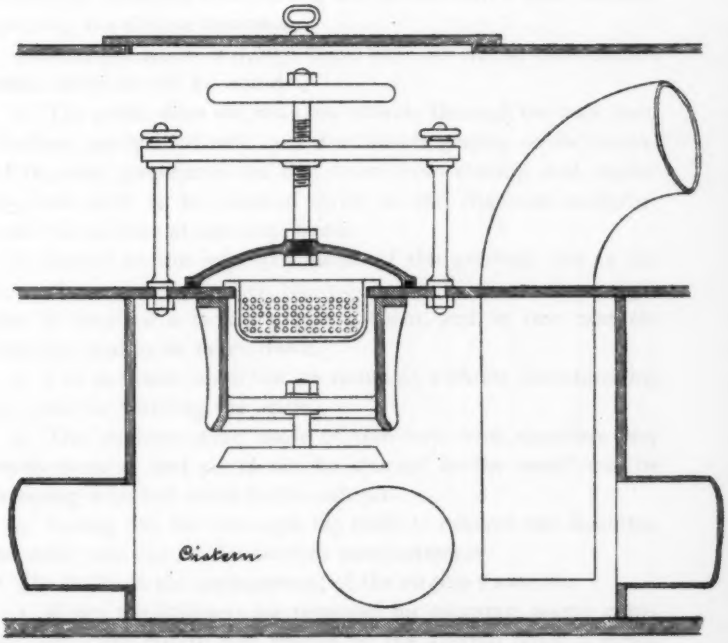


Fig 2

4. The usual position of the ship being "on even keel," the small middle strainer is generally sufficient to keep the bilge dry, consequently the men in charge will allow the gear for working the other valves to rust and stick fast unless periodically reminded to work it.

5. The non-return valves can not be removed, if thought desirable for example to wash out the drains with a hose, without entering the double bottoms.

Besides the faults in design there were several in construction which may as well be noted.

a. The guide stem did not pass entirely through the ball float; the hole partly filled with coal dust which, getting under the end of the stem, prevented the disc valve from closing and caused the ball valve to be jammed down in the chamber, rendering both inoperative at the same time.

b. Owing to the improper shape of the guiding ribs in the non-return valve chamber, the wooden spheres stuck so tight that it required a tackle to move them, and in one case the chamber had to be taken down.

c. The strainers could not be removed without disconnecting the gear for working the valve.

d. The strainers were made of thin cast iron, therefore very easily broken, and could not be cleaned in the usual way by covering with hot ashes in the ash pits.

e. Taking out the through tap bolts to remove the strainers, let water into the double-bottom compartments.

The faults in the arrangement of the engine room are:

1. When the strainers are removed for cleaning, waste, chips, &c., enter the cistern and thence to the suction pipes, a scrubbing brush having passed this way on one occasion.

2. Waste, &c., entering as described, is liable to clog the non-return valve and prevent its action.

3. The accumulation of water in the bilge greatly adds to the difficulty of removing the strainer for cleaning.

4. The non-return valve being only accessible for inspection by entering the cistern through the compartments of the double bottom, is apt to be left to look out for itself a good deal. It

happened on one ship to be in the way of the fireman who cleaned the cistern, so he removed it altogether, and the fact was not discovered for a considerable time.

5. The valve over the strainer, being much in the way of the man who keeps the strainer clear, and being easily removed, is often laid one side, the rolling of the ship carries it over to the lee side of the engine room, the next man coming along stows it in the shaft alley, and finally, on arriving in port and cleaning up for inspection, it is stowed safely away in some remote corner, there to remain until unearthed by an officer and restored to place, only to repeat its travels at some future time.

6. If much water was backing up into the compartment it would be difficult, if not impossible, to close this valve.

7. All parts being below the floor plates might be mislaid for some time without the fact being observed.

8. The manhole doors opening into the double bottoms are liable to leak and let greasy water into these parts, quickly destroying the paint.

To overcome all the difficulties before enumerated I propose in every case to build up above the inner skin a box, as shown in Fig. 3, provided with a water-tight top, the top to be level with the floor and form part of it, and to be hinged and provided with suitable fastenings. Inside of this box is the movable strainer made of galvanized wrought iron, with holes not less than $\frac{1}{4}$ inch diameter. Against the openings in any convenient side is placed the non-return valve. A flat piece of rubber would probably be as efficient as any other. Outside, covering the same openings, is a slide valve operated from a position at some distance above the floor by a system of levers alone, without screws if practicable, the latter being more difficult to keep in working order. The valve moves horizontally in grooves so arranged that the edges of the valve will clear away coal dust or other obstructions liable to lodge in them. Amidships, over the keel, a keelson about 12 inches deep should be made to extend the whole length of the engine and fire room to check the rush of bilge water back and forth across the ship in a seaway. Also, when the floor of the ship is so flat that water

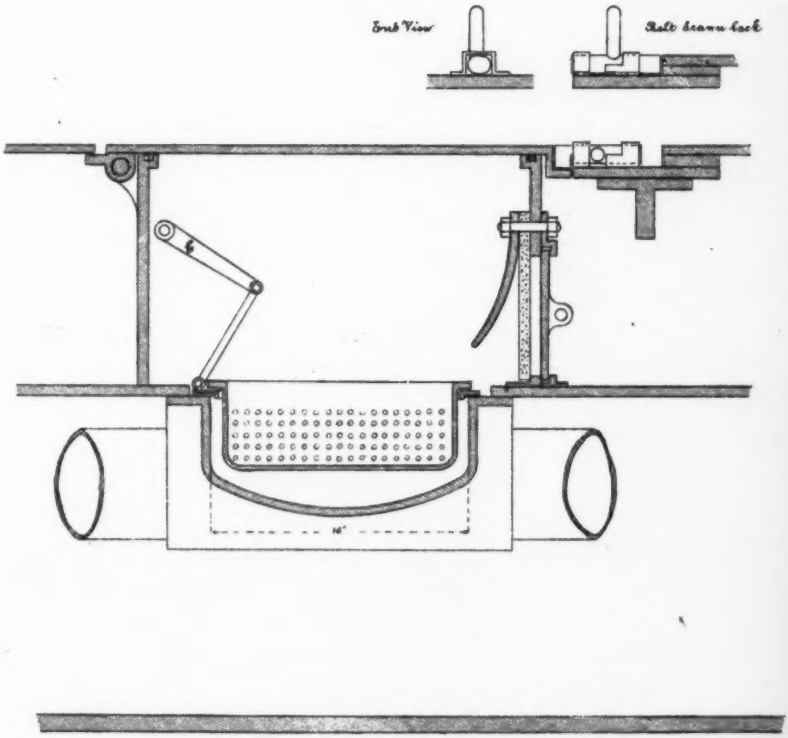


Fig 3

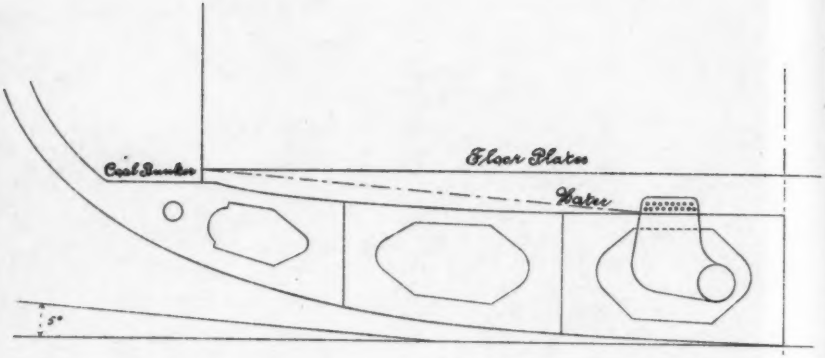

Coal Bunker
Floor Plates
Water
5°

Fig 4

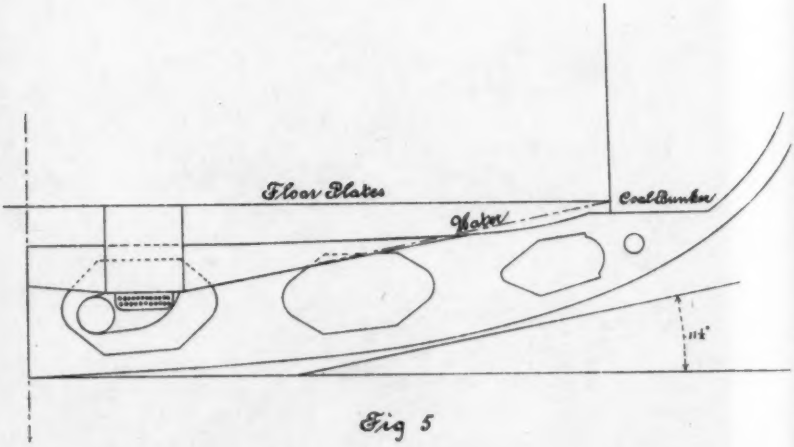

Floor Plates
Coal Bunker
Water
11°

Fig 5

lodges on the lee side with a moderate heel, say 5°, provision should be made for draining it off. This could be done by shaping the inner skin between two frames in the wake of the strainers, so as to form a gutter as shown in Fig. 4, with the amidship end as much lower than the outboard end as might be found practicable. In Fig. 4, drawn to scale, the lines show the strainer as actually fitted and the level of water with the ship heeled 5°, just reaching the floor plates at the bunker door. In Fig. 5 the lines show the strainers fitted as proposed. In this case the water just reaches the floor plates at the bunker doors when the ship is heeled full 11°.

NOTE.—The amidship section used in the sketch is the nearest one attainable to the section at the strainers. The figures are therefore not perfectly exact, but are near enough for all practical purposes.—W. F. W.

The necessity for some such provision for drainage, was made evident when the coal heavers were obliged to wade through 6 inches of water in the lee bunkers or else use all coal from the weather side.

The operation of the proposed arrangement would be: Shut the slide valve, open the box, replace the strainer with a clean one, shut the box and open the valve.

The advantages are:

1. All working parts are easily accessible.

2. The non-return valve is seen every time the strainers are cleaned or changed.

3. The strainer can be changed just as easily when the bilges are full of water as at any other time.

4. The firemen would naturally shut the slide valve without special orders, as stopping the rush of water would lighten his work of cleaning the strainer, consequently waste, &c., would not enter the drains, the mechanism would be kept in working order, and the men familiar with its use.

5. The supplementary strainer being done away with, it is necessary to keep the main valves in working order.

6. All strainers in engine and fire rooms are interchangeable, the box being adapted for use with or without the cisterns.

7. When used in connection with a cistern, the latter can be inspected, entered and cleaned, even if there is considerable water in the bilge, through the opening for the strainer, which is made not less than 16 inches in diameter for this purpose. This is an important advantage in port, as the work of cleaning the cistern can go on while washing out or testing boilers, draining condensers, overhauling the water ends of pumps, &c., or when the steam is shut off the ship or for other reasons it is not convenient to keep the bilge pump going.

8. The manhole doors opening into the double-bottom compartments can be dispensed with.

9. The non-return valve being above the level of the inner skin, the large cast-iron chamber is not required inside the cistern, leaving more room for entering and cleaning.

To preserve the continuity of the double-bottom system of the ship, it is important that the top of the box should always be kept shut and fastened except for the short times it is opened to change the strainers. This is provided for by making the top of the box on a level with the floor plates so that it could not be left open without the fact being quickly discovered.

It is not difficult to contrive a fastening which would project up and be in the way except when secured. A simple fastening of this kind is shown in Fig. 3. It only differs from an ordinary door bolt in that the end of the bolt is made eccentric so that although all parts fit loosely, yet, when the bolt is shot and the handle turned down, the top of the box will be wedged down by the eccentric, and also the handle cannot be turned down when the bolt is drawn back.

It having been suggested that an emergency might arise in which it might be thought advisable to dislodge a strainer when water was above the floor plates, an arm, *b*, Fig. 3, on a shaft passing through a stuffing box in the side of the box, worked by levers from any desired position, has been provided. The arm is connected with a ring in which, ordinarily, the strainer rests, fitting loosely.

NOTES.

THE ATLANTIC LINER, PAST, PRESENT AND FUTURE.

Under the above title, "Engineering" for December 4, 1891, gave some very interesting data of famous Atlantic greyhounds. An abstract is here given:

The White Star steamers *Teutonic* and *Majestic* have beaten the Inman liners *City of Paris* and *City of New York*. The *Teutonic* holds the record each way. The following tables show the best passage of each of these ships:

QUEENSTOWN TO NEW YORK.

Steamer.	Date.	Passage.	Miles steamed.	Average speed in knots.
		d. h. m.		
Teutonic	Aug., 1891.	5 16 31	2,778	20.35*
Majestic	July, 1891.	5 18 8	2,777	20.11†
City of Paris	Aug., 1889.	5 19 18	2,788	20.01‡
City of New York	Oct., 1890.	5 21 19	2,775	19.63§

* Daily runs: 460, 496, 505, 510, 517, 290. † Daily runs: 470, 501, 497, 501, 491, 317. ‡ Daily runs: 432, 493, 502, 506, 509, 346. § Daily runs: 437, 460, 498, 495, 491, 394.

NEW YORK TO QUEENSTOWN.

Steamer.	Date.	Passage.	Miles steamed.	Average speed in knots.
		d. h. m.		
Teutonic	Oct., 1891.	5 21 3	2,790	19.79*
Majestic	Sept., 1890.	5 23 16	2,809	19.61
City of Paris	Dec., 1889.	5 22 50	2,784	19.49
City of New York	Sept., 1891.	5 22 50	2,782	19.44

* Daily runs: 483, 468, 468, 460, 440, 457, 14.

Notwithstanding these high speeds, vessels are now building to do even better, the Cunard Company having ordered two

from the Fairfield Company. The Inman Company are also contemplating some new vessels, but no contract has been made.

"As to the size of the steamship of the future, financial considerations must determine it. Any addition to size means greater displacement and greater power to get a high rate of speed. A small craft, similar, perhaps, to a torpedo boat of the Thornycroft or Yarrow build would give results satisfactory at least on the debit side of the ledger. But what of the credit side, which, after all, is the more important in a merchant ship of the Atlantic greyhound type? In the case of British ships, there are only two sources of revenue—from passengers and for the transport of cargo and mails. In a vessel of the torpedo type there is not room for either. So also with high speed cruisers, where a very large proportion of the length, in some cases 40 per cent., is given up for machinery. Indeed, only in exceptional cases could a cruiser carry fuel to cross the Atlantic at full speed, and certainly no torpedo boat would essay to do so. In the modern high-speed passenger steamer the cargo capacity of the vessel is limited, and so valuable that only prescious cargo is carried, and that at a pretty high rate. The cause of the limitation is the weight of machinery and coal and the necessity for minimizing the displacement. To compensate for this decreased cargo capacity, and to meet the increased expense, it is necessary to secure a larger income, and it may be taken that this can be done more effectually by adding to the passenger accommodation rather than by raising the passenger fares. This means a great increase in the size of the vessel, for passengers cannot well be put below the level of the loadwater line. It is not necessary to enter into the relationship of power, speed and size of vessels. Atlantic vessels are built to comply with conditions dictated by financial considerations."

A very interesting set of diagrams is given, showing that in general the increase of speed has been accompanied by an increase of size, and in general the fastest ships are the longest; thus, the *Teutonic* and *Majestic* though somewhat less in displacement than the fast Inman vessels, are longer.

The engines of the *City of Paris*, indicating 20,000 I.H.P.,

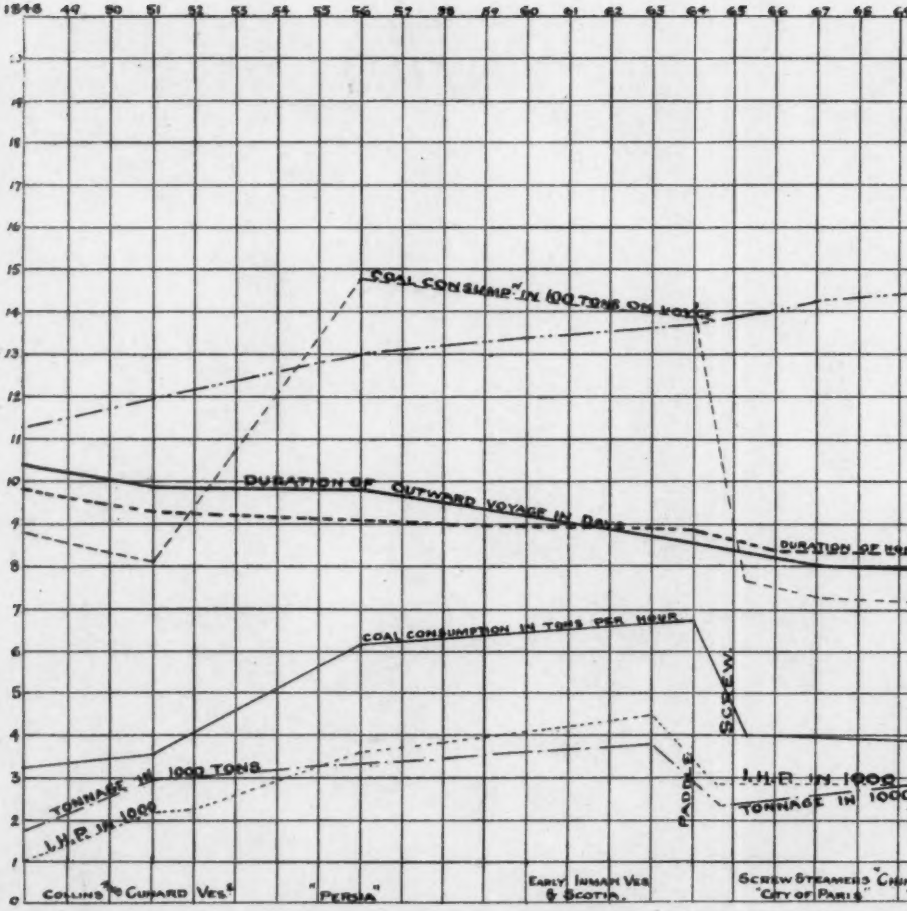
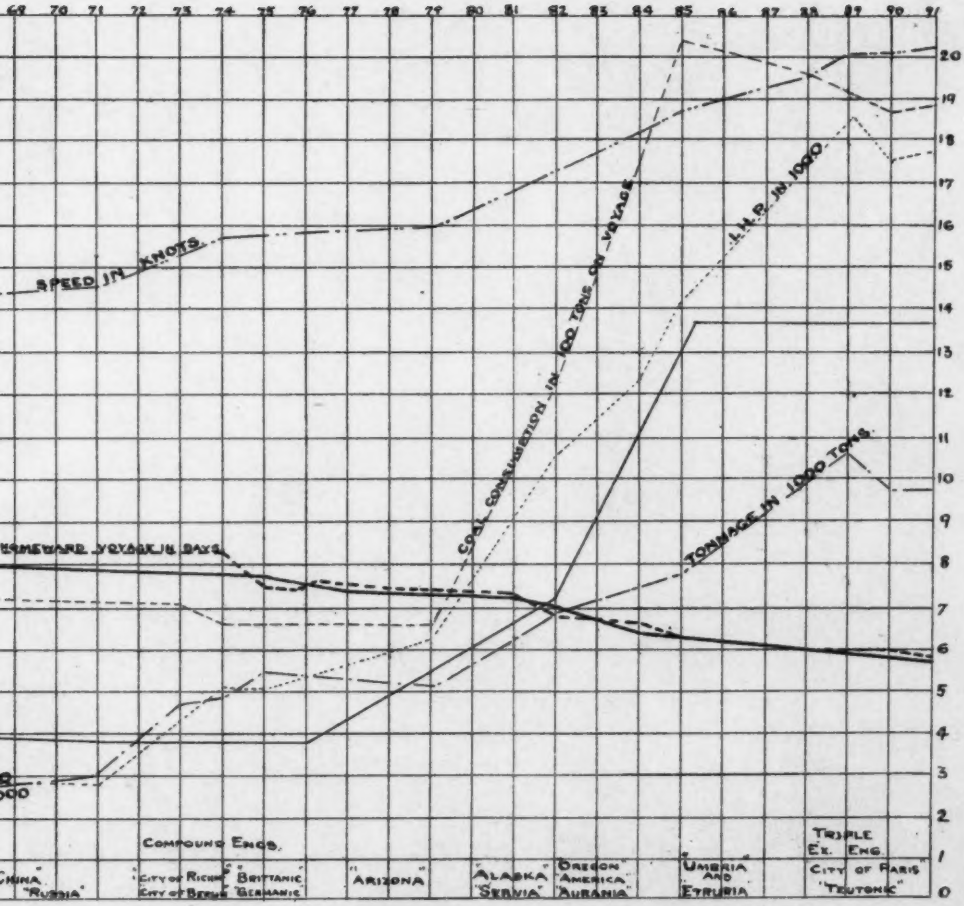

1848 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68
COAL CONSUMPTION IN 100 TONS PER VOYAGE
DURATION OF OUTWARD VOYAGE IN DAYS
COAL CONSUMPTION IN TONS PER HOUR
SCREW
PADDLE
TONNAGE IN 1000 TONS
I.H.P. IN 1000
I.H.P. IN 1000
TONNAGE IN 1000
COLLINS & CUNARD VES.S
"PERSIA"
EARLY INMAN VES.S & SCOTIA.
SCREW STEAMERS "CHI
"CITY OF PARIS"

Diagram illustrating the perform



nance of Atlantic Liners, 1840 to 1891.

weigh with boilers, water in same and in condensers, 2,600 tons, while the engines of the pioneer vessel of the Cunard fleet, the *Britannia*, of 740 I.H.P., were only a fraction of this, probably not more than a tenth. The I.H.P. per ton is, in the *City of Paris* 7.7 and in the *Britannia* about 3.5 or less. Although the first Cunarder would look diminutive beside the later craft, the newer vessel can carry very little additional cargo. The *City of Paris* requires to have 2,500 tons of coal in her bunkers against 600 in the *Britannia*, and that notwithstanding the former does not take half as long to cross as the latter.

The difference financially is made up in passenger accommodation. The *City of Paris* has carried 1,500, whereas the *Britannia* only took 115. In one of the leading Atlantic liners which has been mentioned, 80 per cent. of the income is derived from passenger service alone.

The leading builders do not consider that the limit of size has been reached, and, in fact, the present limit is in the docks; Mr. James Deas, harbor engineer at Glasgow, sets the example by constructing a graving dock with a floor 900 feet long, with a width at bottom of 74 feet and at entrance of 85 feet, and capable of taking in a vessel of 26 feet draught.

"As to speed, there is really no insurmountable difficulty in attaining 40 knots, but this would require something like 160,000 I.H.P., seventy boilers to generate the steam, and these would burn considerably more than 2,000 tons of coal per day. Some idea may be formed as to the size of the vessel necessary for this machinery. We leave it to the imagination of the reader."

The table given herewith of notable Atlantic steamers is very interesting, but it should be remarked that the displacements given are not always correct for the load draught of water. When it was practicable to ascertain the correct figures, they have been given in addition.

The diagram showing graphically the progress in fifty years, is of great interest. The article states that the line "tonnage in 1,000 tons" is the gross tonnage, as it was not practicable to secure the displacement in all cases.

The *Etruria* is practically the same as the *Umbria*, the *City of Paris* is the same as the *City of New York*, so also are the *Teutonic* and *Majestic*, and the *Spree* and *Havel*. The differences in the case of each pair are not important.

BLANCA ENCALADA.

Advertisements have been issued by the Chilean government inviting tenders for raising and repairing the Chilean ironclad, *Blanca Encalada*, which lies almost north and south in the Bay of Caldera, obstructing the anchorage. Her deck is at an angle of ten degrees with the vertical. She lies on her starboard side, resting partly on the damaged portion of her hull. As far as the divers have been able to ascertain, the hole is near the engine-room, and is about 23 feet in length by 10 to 14 in width, and of very irregular form. However, the lower part of the hole is clearly seen, and is to be found about four feet in the perpendicular over the line of the keel. It is difficult of access, owing to the vessel being embedded amidships three feet in the firm sand. With the exception of this hole the divers have not been able to detect any other damage. The size and position of the hole in her hull will render it necessary to take her out of the water altogether to carry out the proposed repairs. Depth of water at high tide, eight and a half fathoms; at low tide, eight fathoms approximately. Tenders are to include the repairs which may be necessary to render the vessel thoroughly seaworthy for three years, so that she may not require docking during that time, and these repairs must be carried out under the superintendence and to the satisfaction of a technical commission to be appointed by the Government. A deposit of $50,000, Chilean currency, will be required as a guarantee for the due completion of the work, from the date on which the order is given to commence work. One hundred and fifty running days will be allowed for the completion of the work. Should the work not be completed within the term named, a deduction of $1,000, Chilean currency, per day occupied in excess of the stipulated time, will be made from the

X

DIMENSIONS, &c., OF SOME NOTABLE ATLANT

Steamer's name.	Builders.	Date.	Dimensions.			Proportion of length to		Draught on trial.	Trial displacement.	Load draught.	Load displacement.	Gross tonnage.	
			Length.	Breadth.	Depth.	Beam	Depth.						
			ft.	*ft. ins.*	*ft. ins.*			*ft. ins*	*tons.*	*ft. ins.*	*tons.*		
Great Eastern	Scott Russell	1858	680	82–6	58–2	8.23	11 7	25–6	27,000	...	...	24,360	....
Britannia	Harland & Wolff	1874	455	46–0	34–0	9.89	13 38	23–7	8,500	...	...	5,004	(2)
Arizona	Fairfield Company	1879	450	45–2	37–6	9.96	12.00	18–9	6,415	26–0	9,000	5,147	62
Servia	J. and G. Thomson	1881	515	52–0	40–9	9.90	12.62	23–3½	9,900	26–6	12,300	7,392	72
Alaska	Fairfield Company	1881	500	50–0	39–7	10.0	12.63	21–0	...	26–0	9,500	6,932	68
City of Rome	Barrow Company	1881	546	52–0	38–9	10.5	9.29	21–6	11,230	26–0	13,300	8,141	(3)
Aurania	J. and G. Thomson	1882	470	57–0	39–0	8.24	12 05	...	...	...	...	7,269	68
Oregon	Fairfield Company	1883	500	54–0	39–9	9.25	12 57	23–8	...	26–6	12,500	7,375	70
America	J. and G. Thomson	1884	432	51–0	37–6	8.47	11.52	26–6	9,300	...	...	6,500	63
Umbria	Fairfield Company	1884	500	57–6	38–2	8 76	13.13	...	10,500	26–6	13,300	7 718	71
Lahn	Fairfield Company	1887	465	49–0	36–6	9.48	12.83	22–0	7,700	26–6	9,500	5,661	(2)
City of Paris	J. and G Thomson	1888	560	63–0	41–0	8.89	13.02	24–6	13,000	26–6	14,500	10,499	(2)
Augusta Victoria	Vulcan Co., Stettin	1889	480	56–0	38–0	8.6	11.65	22–9	9,500	...	...	7,661	(2)
Columbia	Laird Bros	1889	480	56–0	38–0	8.6	11 65	22–9	9,500	...	...	7,578	(2)
Teutonic	Harland & Wolff	1890	520	57–6	39–4	10.1	9.8	26–0	12,000	...	...	9,686	(2)
Normania	Fairfield Company	1890	520	57–3	38–0	9.08	13.7	22–0	10,500	...	...	8,716	(2)
Spree	Vulcan Co., Stettin	1890	485	52–0	38–0	9.33	12.8	22–0	8,900	...	...	6,963	(2)
Fuerst Bismarck	Vulcan Co., Stettin	1891	502½	57–6	38–0	8.73	13.2	22–6	10,200	...	...	8,000	(2)

* The introduction of Howden's system of forced draft has resulted in the shortening of the grate bars by 13 inches, making the new g
of forced draft. † About.

TABLE ATLANTIC STEAMERS.

Load displacement.	Gross tonnage.	Cylinders. Diameters in inches.	Stroke in inches.	Boilers. Heating surface.	Grate surface.	Working pressure.	Indicated horse-power.	Speed on trial.	$\frac{D^3 \times S^2}{I.H.P.}$
tons.									
...	24,360	.....	...	...	...	...	7,650	14.50	359
...	5,004	(2) 48 ; (2) 83.....	60	...	...	70	5,500	16.0	312
9,000	5,147	62 ; (2) 90.....	66	19,500	780	90	6,300	17 0	...
12,300	7,392	72 ; (2) 100.....	78	27,483	1,014	90	10,300	16 9	216
9,500	6,932	68 ; (2) 100.	72	...	972	100	10,500	18.0	...
13,300	8,141	(3) 46 ; (3) 86.....	72	29,286	1,398	90	11,890	18.23	256
...	7,269	68 ; (2) 91.....	72	23,284	1,001	90	8,500	17 75	...
12,500	7,375	70 ; (2) 104.....	72	38,047	1,428	110	13,500	18.3	...
...	6,500	63 ; (2) 91	66	22,750	882	95	7,354	17.8	339
13,300	7 718	71 ; (2) 105.....	72	38,823	1,638	110	14,321	19.0	230
9,500	5,661	(2) 32½ ; 68 ; (2) 85.....	72	...	...	150	9,500	18.64	266
14,500	10,499	(2) 45 ; (2) 71 ; (2) 113.....	60	50,265	1,293*	150	20,605	21.8	278
...	7,661	(2) 106 3 ; (2) 41.3 ; (2) 66.96.....	63	36,000	1,120	150	14,110	18.31	...
...	7,578	(2) 41 ; (2) 66 ; (2) 101.....	66	34,916	1,226	150	13,680	19.15	...
...	9,686	(2) 43 ; (2) 68 ; (2) 110..... ...	60	40,072	1,154	180	18,000	21.0	269
...	8,716	(2) 40 ; (2) 67 ; (2) 106.....	66	46,490†	1,452†	160	16,352	19.34	212
...	6,963	(2) 38 ; 75 ; (2) 100.....	72	36,000	1,120	165	13,000	19.6	249
...	8,000	(2) 43.37 ; (2) 67 ; (2) 106.25.....	63	47,000	1,450	157	16,412	20.7	254

, making the new grate area 1,026 square feet. The results given above are under the closed stokehold system

contract price; and, on the other hand, should the work be finished before the expiry of the time allowed, a bonus of $1,000, Chilean currency, per day, will be paid to the contractors.

D'ALLEST BOILERS.

An article in "La Marine Française" questions the advisability of putting twelve of these boilers in each of the ships, *Bouvines*, *Jemmapes* and *Valmy*, because of the rather unfavorable experience with them thus far. It states that boilers of this type have not been used for long voyages except on the *Libau*, of the Fraissinet Co., and that there an accident occurred which severely injured several persons.

On the trials of the *Bombe*, however, which has two d'Allest boilers, everything went well. For two hours a speed of about 19.2 knots and a horse-power of about 2,000 was maintained, the engines making from 330 to 340 revolutions. The boilers worked well and showed no bad effects of the forced draft. With the locomotive boilers, previously in use, the maximum speed was from 17.5 to 18 knots, and, after every attempt to exceed this, there was trouble with the boilers, such as collapse of the furnaces, severe leakage, &c.

The article then says: "This is a success for the d'Allest boiler, but only a relative one. Further experience should have been secured by placing the *Bombe* in a squadron in place of the *Dague* or the *Dragonne*, whose boilers are almost worn out. This would have settled the accuracy of the claims of certain officers that the d'Allest boilers do not permit cleaning or the removal of a defective tube without great difficulty. These are serious defects, and would, indeed, be enough to cause the rejection of these boilers, were it not that other tubulous boilers, those of *Du Temple*, *Oriolle*, and *Godard* and *Petit*, have shown the same defects in experiments which are now going on.

DEFECTIVE BOILERS IN ENGLISH WAR VESSELS.

[Editorial in "Army and Navy Register," January 16, 1892.]

For some time past the record of English trial trips has been anything but satisfactory. One ship after another has been

compelled to give up the forced-draft trials on account of trouble with the boilers. It has now culminated in the failure of the *Blake*, which in many respects was the pride of the English Navy, with her expected speed of 22 knots and 20,000-horse-power. It is now stated officially that the boilers will not be forced, and that the speed and power already obtained with an air pressure of less than half an inch—about 19 knots and 14,500 I.H.P.—are the maximum.

It may be interesting in this connection to recall the remarks of Mr. W. H. White, director of naval construction of the English Navy, in connection with the paper of Mr. J. Harvard Biles on our new ships. Mr. White assumed a very patronizing air, which, however, is characteristic of him, and said he would not criticise our ships because some of them were built from his designs. He also said that the estimated sea speeds of our *New York* and *Cruiser No. 12* (the *Pirate*) were too high and "we are more modest in our estimates of such speeds than our friends across the water. We are content to speak of the *Blake* and *Blenheim* as likely to maintain, under working conditions, a speed of 17½ knots in smooth water with clean bottoms, as against 20 knots with natural draft on the measured mile." It is unfortunate that the modesty of the estimate had not extended to the measured mile performance, as it has turned out a knot less than the estimate.

The trouble with the boilers has lasted so long that it is now somewhat difficult to say just when it began, but the first cases which attracted special attention were what is known as the "M" class of cruisers, about the size of the *Raleigh* and *Cincinnati* of our Navy. There are five of these and every one had trouble with the boilers, so that while some, after the several attempts, managed to complete the forced-draft trials, it was only with leaky boilers and the certainty that the performance could never be repeated.

Then came the *Sharpshooter* class, where the experience was even worse, and where the ultimate result was a decision to rest content with 3,500 I.H.P. instead of 4,500, as expected.

The disaster on the *Barracouta* is probably familiar to all our

readers, where, owing to leaky tubes, there was a case of back draft causing the burning of several men and their subsequent death.

We need not continue the list, and will now say a word as to the cause. As long ago as July of 1889, an English service paper had these words: "The cause of all the trouble, worry and anxiety that have been occasioned by the propelling machinery of the 'M' class is not far to seek. It is the extreme lightness of construction, verging on the dangerous, which is rendered necessary by the determination of the shipwrights not to allow sufficient space in the ship for the horse-power they demand from the engineering contractors. It is the boilers that are most in fault. To the insufficiency of heating, and especially of grate surface, most of the trouble may be ascribed."

A recent English technical journal speaking of the *Blake's* trial says: "Unfortunately, the boilers are of the same pattern as those which had developed deplorable weaknesses when tested in other vessels. For these failures the contractors were in no way responsible, seeing that they had strictly complied with Admiralty specifications. With the object of economizing weight and space, the boilers were placed in cramped situations, and designed with such limited margins that the men were unable to drive the engines at the highest power without the application of dangerous air pressures for the purpose of promoting rapid evaporation and accelerated generation of steam. But the boilers, because of their faulty design and light construction, were unable to resist the excessive strains set up by this system of torture. The extremely high temperatures produced in the furnaces by the operation of the fans superheated the reduced tube plates to such a degree as to destroy the contact between them and the water, through the violent ebullition established, and led to the rapid deterioration of all the parts exposed to the action of the flames, with leakage from the tubes at the fire-box ends. Double-ended boilers, where six or more furnaces are connected with a common combustion chamber, are also liable to back drafts and chills when working with close stoke holds under great pressure."

Here we have the explanation of the whole trouble—the single combustion chamber for six and eight furnaces, and the very insufficient heating surface. A few instances will show what a very small allowance of heating surface per I.H.P. was provided. In the "M" class it was 1.4 square feet per I.H.P.; in the *Sharpshooter* class, 1.22; and in the *Blake*, if the latest figures are reliable, 1.35. How small these are may be seen by noting that, in the *Baltimore*, where the ratio is the least of any of our new ships, it is 1.63; while in the *San Francisco* it is 2.03, and in the *Philadelphia* 2.32. In the *New York* provision is made for 2.01 square feet per I.H.P., and in *Cruiser No. 12*, for 2.20.

In none of our new ships has the vicious single combustion chamber in double-ended boilers been allowed, as Commodore Melville steadily refused to agree to it, though it was strongly urged for some of our early ships by amateur designers who pointed to the great saving in space and weight attained in the "M" class of the English Navy. It turns out, as was predicted by the engineering experts, that this saving was only on paper, and we can congratulate ourselves that our Engineer-in-Chief had not only the wisdom but the power to prevent such a mistake. Until recently the engineer-in-chief of the British Navy has been little more than a chief draughtsman for Mr. White, whose word was law, and who not only designed the hull but selected the type of machinery and fixed the weights. Now the engineer-in-chief is held responsible entirely, and has the advice of a committee of chief inspectors of machinery. The very first thing done after the inauguration of this change was an increase in the heating surface and weight of all new boilers, and the entire abandonment of the single combustion chamber.

This is a wholesome improvement, and will not only benefit the English Navy, but increase the knowledge of our amateur designers, all of whose wisdom seems to be drawn from a study of English practice.

In connection with this subject the following paragraph from an English service paper is of interest:

"It is now authoritatively stated that the Admiralty have

determined to lift the defective boilers out of the *Vulcan* torpedo ship and to supply her with a new and improved set, by this means enabling her to realize her estimated power and speed. Before this necessary and expedient step was resolved upon, vart ious devices were tried for the purpose of arresting the persisten- leaking of the boilers. A nest of tubes was taken from each boiler to promote superior circulation, and adjustments were made in the furnace grates. It was, however, found that, while there ensued no reduction in the leakage, the resulting decrease in the heating surface materially detracted from the performances of the vessel obtained at the official trials."

SUEZ CANAL PETROLEUM REGULATIONS.

The new regulations for the transit of ships laden with petroleum in bulk through the Suez Canal are as follows:

"1. Any ship laden with petroleum oil in bulk shall, on arriving before any port of access to the canal, at once make herself known by flying at the mizzen one of the signals hereafter described, which shall remain flying during the whole of her transit. Whenever any such ship makes any stay in port she will be isolated by means of floating booms.

"2. Before obtaining entrance into the canal, the captain shall make declaration to the following effect: (1) That his ship is specially classed for the carriage of petroleum oil in bulk in class; (2) that her cargo comprises refined petroleum oil only, of a uniform quality, no sample of which, taken at the port of loading, shall have given a flashing point below 23 degrees Centigrade (73 degrees Fahrenheit), this temperature having been ascertained conformably with such process of close test as may be recognized and made use of in the petroleum oil trade, as for instance the Abel test or any other close test of a not lesser degree of accuracy; (3) that her sole means of lighting, except the regulation apparatus for night transit and her mast lights, are electric incandescent lamps, and that all her electrical installations are in accordance with the rules of Lloyd's or Veritas; (4) that the ship is provided with pumps intended solely for the loading and unloading of petroleum oil, and having an effective

pumping capacity of 300 tons weight of oil per hour at a minimum; (5) that no single tank in the ship has a cubic capacity greater than 200 tons measurement, nor can discharge its contents into any adjoining tank through any aperture or want of continuity whatever of its walls; (6) that no tank is placed either in the bows or stern of the ship; (7) that the portion of the ship occupied by the tanks is completely separated from the forward and after parts by double oil-tight bulkheads, forming spaces wholly filled with water.

"3. Any steamer laden with petroleum oil in bulk must be convoyed during the whole of her transit through the Canal by a tank-tug, constructed with a view to the rapid and immediate lightening of the ship in case of grounding. The company reserves the right of approving beforehand the type and the dimensions of this tug, which shall moreover conform to all the provisions of paragraph 5 of Article 12 of the Regulations for Navigation. The company have a tank-tug, which they will place at the disposal of vessels laden with petroleum in bulk, passing through the canal, whose captains may ask for it. The charge for hire will be in accordance with the provisions of paragraph 3 of Article 12 of the Regulations for Navigation relating to convoying by a first-class tug.

"4. During the whole of the transit through the canal the captain shall take care that no one on board shall smoke, nor have about the person any means of obtaining a light.

"5. Any ship which, after having carried a cargo of petroleum oil in bulk, may require to go through the canal empty of oil, shall not be subject to the provisions of these provisional regulations, but her captain shall make a declaration to the effect that he has carried out all operations of cleansing, cleaning, or others necessary to avoid the presence of any vapor from petroleum oil in tanks or in any other part of the ship.

"6. All the provisions of the Regulations for Navigation, and any appendix thereto not contrary to the provisions of these provisional regulations, remain applicable."

A footnote indicates the signals to be kept flying by the petroleum steamers during the transit through the canal and their

stay in the ports of access, in accordance with Article 1 of these provisional regulations. They are a red flag above one ball by day, and a white light between two red ones by night.

COAL CONSUMPTION IN MEN-OF-WAR.

[Editorial, "London Engineer," January 22, 1892.]

The annual reports of Mr. Melville, chief of the Bureau of Steam Engineering in the United States, always contain useful and interesting information. That for the year 1891, although dated October 12th, has not long been in this country, some time elapsing after it had been written before publication. It forms no exception to the general rule. A large portion of it is devoted to the consideration of the pay and position of engineers, machinists and firemen in the United States Navy, and recommendations to Congress, about all which we may have something to say at another time. For the present we shall confine our attention to a statement made by Mr. Melville, the importance of which, in one sense, can scarcely be overrated. It is simply that ships of war, fitted with the most modern engines and boilers, burn far more coal per horse-power than is commonly believed; and that as a consequence their radii of operation is nothing like so great as those stated in official documents. We shall explain in a moment the nature of the basis of Mr. Melville's contention, but before doing this it is desirable that the bearings of the question should be fully comprehended. It is commonly assumed that triple-expansion engines in the navy will burn at the most about 2 lbs. of coal per indicated horse-power per hour. A ship indicating 10,000 horses-power, and steaming, say, 15 knots, will run 360 knots per day, and burn in round numbers nine tons of coal per hour, or 216 tons in twenty-four hours. If she carries 750 tons of coal, as she dare not leave her bunkers quite empty, she can steam for three days only at full speed, and will run in that time 1,080 miles. Steaming at seven knots she will need about 1,100 horses-power and burn in round numbers 24 tons per day. She will run per day 168 knots, and she can keep the sea for about twenty-seven days and run 4,500 miles. Her radius in the first case would be offi-

cially stated as 1,080 miles, and in the second as 4,500, and various arrangements for re-coaling can be based on such calculations. It will be seen that it would be of the last importance in war time that it should be known precisely how far a ship might venture from a coaling port. Thus, under the circumstances stated, our imaginary ship dare not go further than 500 miles from a coaling port if she had to run at full speed out and home, and it is very easy to see that a mistake in the figures might entail extremely disastrous results.

Now, unfortunately, no experiments pretending to be exact have ever been made on a large scale in this country to ascertain the coal consumption of war ships. Now and then a rough kind of measurement is made during trial trips. The only persons who really know much about the matter are the engineers in charge; and if they have the luck to get a continuous run, say, from Portsmouth to Gibraltar as nearly as possible at the same speed all through, they can arrive pretty clearly at the actual consumption needed to drive their ship at a given speed; but there is still left plenty of room for error. Mr. Melville has taken the bull by the horns, and carried out a series of experiments to settle the question—for the United States Navy, at all events. The results of the trials are sufficiently startling. The vessels tested were the *Concord*, the *Newark* and the *Bennington*. The coal was accurately weighed before-hand into bags containing 110 pounds each, so that it was an easy matter to secure absolutely trustworthy figures. For the *Newark* there was a supplementary coal consumption trial, when two boilers were used with forced draft under conditions as nearly identical as possible with those of the contract trial, the coal being weighed on a platform scale as taken from the bunkers. It was assumed that the coal burned per square foot of grate would be the same in the two cases, and this is probably nearly true.

"These results," says Mr. Melville, "obtained from these accurate trials give the first trustworthy figures in regard to the coal required per horse-power under forced draft full power that have ever been published. They show conclusively what this Bureau has steadily maintained, that the figures so often pub-

lished in estimates of coal endurance are entirely too low, and that the radii of action predicated therefrom are entirely too high. While these results will undoutedly be a source of regret to all who had mapped out fields of usefulness for our vessels based on the more roseate view, there can be no doubt that the best interests of the Government have been served by finding out the truth. It is sure to be brought home to us some day, and it is far better that we should know it now, and act upon it by providing large coal capacity, than wait for the humiliation of a ship found to be helpless when we were most in need of her services."

These ships have triple-expansion horizontal engines. The *Newark* indicated 9,131 horse-power, including all auxiliaries, and burned 2.434 lbs.; the *Concord* indicated 3,513 horses and burned 2.76 lbs.; and the *Bennington* indicated 3,533 horse-power and burned 2.6 lbs. of coal per horse per hour. The coal in the bunkers of the *Newark* was, we are told, semi-bituminous and excellent, the coal for the other two ships was anthracite, and also excellent. The rate of consumption was not very high for forced draft, being about 40 lbs. per square foot per hour. The highest air pressure was 2.45 inches, the lowest 2.25 inches. The trials lasted 4 hours, and were in all respects complete and satisfactory. We see no reason to think that English engineers would do any better, and we believe that consumptions anywhere near 1½ lbs. per horse per hour, with the engines going at full speed exist only in the sanguine imaginations of a few, and we do not believe that the action of the moderate forced draft we use makes any material change for the worse as compared with the results to be had when the fires are pushed as much as possible with natural draft, which, indeed, in a man-of-war, always means assisted draft, the fans running even though the stokehold doors are not shut.

Mr. Melville is not content with making a bald statement. He goes on to show why great economy is not to be expected, and his words are suggestive and instructive. He admits that naval engines are not so economical as those in merchant ships, and goes on: "An interesting comparison of the conditions of the two cases is furnished by the *Baltimore* of our Navy and a mer-

chant steamer called the *Iona*, which have about the same displacement—4,450 tons. The *Baltimore's* machinery develops 10,000 indicated horse-power. The *Baltimore* has room for only 17,000 square feet of heating surface in her boilers with a ratio to grate surface of less than 30, while for the *Iona* the figures are 3,160 and 75. The *Iona* works always at full power, and secures a speed of about 8½ knots; for this same speed the *Baltimore* would require more power on account of the friction of the enormously larger engines. But the great economy is in the boilers. With the enormous amount of heating surface for the power developed the *Iona* can evaporate 10.5 pounds of water per pound of coal, while the *Baltimore* probably does not exceed 8 pounds. The *Baltimore's* boilers weigh 490 tons for 10,000 indicated horse-power, and the *Iona's* 122 tons for 700 indicated horse-power. Were the *Baltimore's* boilers built for economy instead of power, on the same ratio as the *Iona's*, they would weigh 1,743 tons, or nearly twice as much as the entire machinery of the *Baltimore* does. The comparison was made between these two ships purposely, because they are supposed to be of the best English design. In other words, economical machinery means heavy machinery taking up much room; but the power required in swift war vessels of moderate size is so great, that to make the machinery both powerful and economical the whole ship would have to be given up to it. As the great requisite in our ships is powerful but light machinery, economy must of necessity be sacrificed."

No doubt some of our readers will be prepared to traverse Mr. Melville's statements, and to argue that large dimensions are not essential to economy. He has said enough, however, to show that the figures concerning coal consumption which appear in the Naval Annual are probably very misleading; but we have at least the satisfaction of knowing that in all likelihood none of our neighbors are a bit better off than we are.

BASIC BESSEMER STEEL.

The following circular from the Navy Department is regarded as a decided and wise step in the right direction:

"Circular No. 43.—Navy Department, Washington, Nov. 6,

1891.—The clauses which specify the kind of steel in the 'Specifications for Inspection of Steel for use in the Hulls, Boilers, and Machinery of Vessels,' which have been issued by the Navy Department, or which may be hereafter issued, shall allow the use of Basic Bessemer Steel.—B. F. TRACY, *Secretary of the Navy.*"

NEW CLASSIFICATION OF SHIPS.

The following circular will be read with interest:

"U. S. Navy Regulation Circular No. 94.—Navy Department, Washington, D. C., December 26, 1891.—Classification of Ships. —Paragraph 1, Chapter 1, of the Navy Regulations, is amended to read as follows: The ships of the Navy of the United States will be classed as follows: Ships of and above five thousand tons displacement will be classed as *first rates;* those of and above three thousand, but below five thousand, tons displacement, as *second rates;* those of one thousand and above, but below three thousand, tons displacement, as *third rates;* and all those of less than one thousand tons displacement as *fourth rates.* —B. F. TRACY, *Secretary of the Navy.*"

INSPECTORS OF MACHINERY AFLOAT IN THE BRITISH NAVY.

Quite important changes in the Engineer Department of the English Navy have been introduced since the appointment of Admiral Hoskins as Senior Sea Lord of the Admiralty.

For many years the fleet engineer of a squadron was also the inspector of machinery afloat, but as the squadrons have grown in strength and number and the engines in each ship have increased in complexity, it is considered that this arrangement is no longer practicable, and that the ordinary duties of a large ship are quite as much as can be efficiently discharged by the chief engineer.

The intention of the new lord is to attach an inspector of machinery to every squadron in addition to those who are stationed at the naval establishments at home and abroad. The new arrangement will give a little welcome promotion to the members of the Naval Engineer Corps. The inspector of machinery will be a member of the admiral's staff of each squadron.

SHIPS.

UNITED STATES.

Baltimore.—The deterioration of the condenser tubes of the *Baltimore* and the necessity of the replacement of the entire lot has called attention to a phenomenon which is certainly curious, and for which thus far there is no explanation which has received general acceptance, or is even entirely plausible.

The *Baltimore's* condenser is of English design and carries out the English naval practice of having the steam inside of the tubes, and the refrigerating water outside. Owing to a variety of causes, the space around one end of these tubes became almost solid with mud, and when the vessel arrived at Mare Island, Cal., recently, advantage was taken of the opportunity to remove the tubes and thoroughly clean out the condenser.

On removing the tubes there was nothing in their appearance to indicate that they had deteriorated, but it was found that they broke short across by a very slight blow. This led to a careful examination, and it was found that all the tubes were in this same condition. They could be broken across the knee as easily as a stick. The fracture showed a complete change in the material. A thin ring on the inside had the color and appearance of the brass of which the tubes were originally composed, but, outside of this, the rest of the tube was of a dull copper color, without metallic luster, and giving the impression of fine particles of some material deposited from a solution.

The whole phenomenon is something so entirely different from the usual experience with condenser tubes, which have generally been considered almost indestructible with decent treatment, that an explanation seemed impossible.

One theory was that the deterioration was caused by having steam inside of the tubes instead of outside as has always been the custom in our Navy, but no such occurrence has been reported with English condensers built in this way, and inquiry

revealed the fact of such a condenser in an American merchant vessel, whose tubes after fourteen (14) years of service were as good as when put in.

Another theory is that, as no such trouble was ever experienced while the shells of the condensers were of cast iron, the deterioration is consequent upon the use of brass condenser shells, the idea being that the tubes and the shell are sufficiently far apart on the electrical scale to form a battery in which the tubes are the positive pole and are attacked, the zinc in them being dissolved out.

This last theory would seem quite plausible were it not for the fact that copper pipes on the *Baltimore* and some other ships, connected with the condenser, have also shown marked deterioration.

As far as can be learned there has been trouble on nearly all of the new vessels with copper pipes, and it is learned that this is not confined to our Navy, but has appeared upon ships of the British Navy.

Altogether the matter is a most interesting one, and a correct explanation will be awaited with great interest. The chemical analysis of some of the defective tubes of the *Baltimore* is now going on and may throw some light on the question.

In addition to the foregoing the following extracts from a recent report of a chemical analysis made by Assistant Engineer Andrew M. Hunt, U. S. Navy, will prove of interest and value. He says:

"I first examined a large number of tubes. In all cases I found the tin coating on the inside of the tubes bright and intact. The outside surfaces were dull in appearance, and rubbing them down with canvas, I found the tin coating entirely gone in spots. Some tubes had hardly any tin on the outside surfaces."

Wherever the tin coating remained the metal was unchanged beneath, but in the places from which the tin had been removed the metal of the tube was coppery in hue, and so brittle as to be easily broken in the fingers. A microscopic examination showed the fractured metal to be porous and spongy, while the yellow or uninjured metal was tough and unchanged.

There was no shading off of one color into the other, but the line was sharp and clearly defined.

Chemical analysis showed that the reddish-brown metal was nearly pure copper, not a trace of zinc being shown.

The yellowish metal was brass, of a composition of copper, 60 per cent.; zinc, 40 per cent., and samples taken as close to the red metal as possible showed no material variation from these figures.

Chemical analysis of metal from spare new tubes demonstrated the equality of the tubes in the condensers with those in store.

The action by which the zinc was removed from the metal of the tubes evidently progressed from the outside of the tubes toward the inside, as evidenced by the unbroken tin coating inside, and the fact that numerous samples were found where copper of the outside metal remained with an interior shell of brass.

The complete removal of the zinc from the alloy indicates galvanic action, while the sharp lines of demarcation of the altered portions and the way in which the alteration has spread from centers of action, indicates that the galvanic action had taken place between different parts of the same tube, and not between the metal of the tube and some outside body.

In explanation of the facts as ascertained Mr. Hunt says: "To have accumulated so much mud in the condensers, a very large quantity of it must have been swept through, and, containing a considerable quantity of grit, it would abrade the tinned surface of the tubes. This abrasion would be rendered more rapid owing to the fact that the tin of coating contained such a large per centage of lead as to be quite soft. The fact that the tubes were more affected, and tin coating most removed, at points where the currents of water moved transversely to the tubes, and that the tubes were least affected where the deposited mud protected them from the abrading action, renders this explanation of the manner of the removal of the tin coating extremely probable. Where the tin coating had been removed at any one point there was then left a brass surface exposed, adjacent to and intimately connected with a (tin lead alloy) surface, and covered by exciting liquid, sea water. * * * If the above theory is

correct, had the tin coating on the tubes remained intact, no such action could have taken place, and I am also of the opinion that had the tubes not been tinned at all, they would not have been affected. As the galvanic action, in all probability, took place between different elements in the same tube, I doubt if fitting zinc slabs will in any way prevent a recurrence of the trouble if the tin coating is again removed from the tubes."

New York.—The *New York*, heretofore known as *Armored Cruiser No. 2*, was launched at Cramp's Shipbuilding Works, Philadelphia, December 2, 1891, and the work of fitting the vessel for sea is progressing as rapidly as is consistent with proper attention to details.

Some changes have been made in the minor fittings of the ship, but no notable deviation from the original plans has been found necessary.

Machias.—December 8th, 1891, at the yard of the Bath Iron Works, Bath, Maine, the *Machias*, better known as *Gunboat No. 5*, was launched.

The date of her completion, ready for commission, is somewhat uncertain, but work is being pushed with all dispatch.

Montgomery.—On the 5th of December last, *Cruiser No. 9* was launched from the yard of the Columbian Iron Works, Baltimore, Md., and is being rapidly prepared for commission.

The name of *Montgomery* was given to this cruiser out of compliment to Chairman Herbert, of the House Naval Committee.

Detroit.—*Cruiser No. 10*, under construction by the Columbian Iron Works, of Baltimore, Md, was launched on the 28th of December, 1891, and on the same occasion was named the *Detroit.*

ENGLAND.

Torpedo Boats.—Mr. A. F. Yarrow, having been asked to estimate the rate of speed with which, under pressure, he could construct first-class torpedo boats similar to those boats of his which were added to the Navy in 1889, has replied that to finish six boats he should require four months, and to finish twenty, eight months. This declaration disposes of the theory that Great Britain may, with comparative safety, delay the building of

further torpedo boats until a probability arises that such craft will be at once required; and it lends significance to a recently published official French statement of the number of first-class torpedo boats actually belonging to the Great Powers. The numbers are: France, 184; Italy, 123; Great Britain, 85; Germany, 79; Austria, 57; and Russia, 32. France, Italy, and Germany have not for some time ceased to lay down torpedo boats, and are very rapidly adding to their flotillas. In this country the boats which were delivered by Mr. Yarrow more than two years ago were the last to be laid down for the Royal Navy.

The Second-class Cruiser *Forte*.—The stem has been cast at Sheerness Dockyard for a new second class cruiser, named the *Forte*, which is to be built for the Royal Navy at Chatham Dockyard. The *Forte* is to be built upon similar lines to the *Apollo* class, with the exception that she will have a greater length, breadth and displacement. She will be 320 feet in length, and will displace 4,360 tons. The engines of the *Forte* are to be of 9,000 horse-power, and she will carry an equipment of two 6-inch, eight 4.7-inch and nine 3-pounder quick-firing guns. Eight cruisers of the *Forte* class are to be built for the Royal Navy under the Naval Defence Act.

H. M. S. *Vulcan*.—Paradoxical as it may seem, the *new* torpedo depot ship *Vulcan* has grown *old* and practically worn out, before ever having been commissioned, and she will shortly be sent to the Thames from Portsmouth, there to have her boilers renewed and her attenuated hull strengthened. The *Vulcan* was specially designed to accompany a fleet with a large equipment of torpedo boats, mining stores, &c., and is fitted up as a floating workshop. Her first steel plate was laid on the 18th of June, 1888, and she was launched on the 13th of the same month in the following year. The better part of two years was thereafter spent in preparing her for sea, during which time considerable structural alterations were made and new ports were cut for submerged torpedoes. In April of last year she went on her official eight hours' natural-draft trial with satisfactory results. During the trial the speed realized over the measured mile was 18.59

knots, while the mean H.P. developed over the eight hours was 8,160, or about 900 in excess of the contract limit. Subsequent essays, however, under forced draft were otherwise than gratifying, experiment after experiment being made without avail, each succeeding trial, indeed, being if anything less successful than the one preceding, so that the original anticipations of 20 knots being realized with 12,000 H.P. have never been approached. In fact, latterly a speed of 16 knots could scarcely be conceded without the unwelcome accompaniment of leaky boiler tubes. Such is the brief history of the *Vulcan*. She is certainly the first of her kind, and may be looked upon as an experiment, although up to date a costly one and a failure. Her boilers are primarily to blame, and after much experimenting the Admiralty have been forced to the conclusion that the only remedy is to replace the present double-ended boilers with common combustion chambers by others of the old type with the usual combustion chambers. This will have to be done at a serious expenditure of money, but it is to be hoped that when re-boilered and otherwise resuscitated, the *Vulcan* may more closely approach the original intentions of her designers.—*Marine Engineer.*

Fox.—The *Fox* is to be a second-class cruiser, following, in a general way, the model of others of the class, but will vary somewhat in important particulars.

The length of the *Fox* is 320 feet, while the *Latona, Naiad* and others are but 300 feet.

Her beam will be 50 feet as against 43 feet 6 inches for the others, and in displacement she will exceed those mentioned by nearly a thousand tons, giving considerably more space for engines and boilers.

The *Fox* will have engines of 9,000 I.H.P., about the same as the rest of her class. The boilers will show a marked departure, as regards efficiency, from those furnished other ships of her class, the authorities having determined on abandoning the locomotive type for a boiler more fitted to the requirements of a sea-going cruiser.

To quote the "Naval and Military Record," under date of January 7, 1872: "There will be no mistake either by the Ad-

miralty in supplying her (the *Fox*), or, for that matter, any more of the new ships to be taken in hand for the future, with that vile model of the locomotive boiler that has already played such havoc with our new ventures afloat; for, thanks to common sense overcoming official obstinacy, the double-ender fad of the Whitehall experts has at last received its quietus, and the order been issued that no more boilers of this pattern are to be supplied to the ships of the Navy."

The *Fox* will be fitted with a flush deck running fore and aft, in place of the poop, forecastle and waist of her prototypes. Her freeboard will be higher, and her gun platforms will be raised several feet higher than in others of her class.

Indefatigable.—The *Indefatigable* is a second-class cruiser of 3,500 tons displacement, built and engined by the London and Glasgow Engineering and Shipbuilding Co., was subjected to a full-power steam trial under natural draft, on January 19.

The following results were obtained:

Steam pressure, 134.7; vacuum, 27 and 26.9; revolutions, 126 and 127.5; I.H.P., 3,702 starboard, 3,619 port, an aggregate for the main engines of 7,321. The contract was for 7,000 I.H.P. The speed, by patent log, was 18.74 knots. Three days afterward, January 22, the forced-draft trial was attempted, but owing to want of sufficient air space between the grate bars, preventing the consumption of coal with the proper rapidity, the trial was abandoned, there being no prospect of the guaranteed power being developed.

Gleaner.—This vessel, a torpedo gunboat of 450 tons displacement, one of the *Sharpshooter* class, was given a trial in the latter part of December, and secured about 3,630 I.H.P. and a speed of 20.1 knots for three hours; the steam pressure was 147 lbs.; mean revolutions, 249.6; and vacuum, 25.5 inches. This was considered as exceeding the required I.H.P. of 3,500, but this had been "scaled down" from 4,500, which was the original estimate of the *Sharpshooter* class. As none of them have ever much exceeded 3,500, that is now regarded as the estimated I.H.P. An English Service paper says: "The *Gleaner* may possibly be able to overtake the majority of our own torpedo

boats—unless, like her sisters, she should break down in service—but neither she nor any of her class can be depended upon to capture or destroy the fast boats of our neighbors across the Channel. The French will shortly possess 200 first rate torpedo boats."

Blake.—Notices of this fine ship and her sister, the *Blenheim*, have been given, but, to bring all the information together, the dimensions will be repeated. The hull is 375 feet by 65 feet by 25 feet 9 inches, with a displacement of 9,000 tons. There are two sets of triple-expansion engines to each shaft, with cylinders 36, 52 and 80 inches diameter by 48 inches stroke. Each set of engines has an air pump 33 inches in diameter and 2 feet stroke, and a condenser with 3,200 tubes and 5,625 square feet of cooling surface, the aggregate of the four being 22,500 square feet. Each condenser has an independent centrifugal circulating pump with a runner 3 feet 9 inches diameter. The propellers are 18 feet 3 inches diameter and 24 feet 6 inches mean pitch.

There are six main boilers, double-ended, 18 feet long and 15 feet 2 inches diameter, with four furnaces at each end, the whole eight connecting to a single combustion chamber. Owing to trouble with similar boilers on other ships, an effort was made to avoid it in these by building in brick partitions. There is one auxiliary boiler 9 feet long and 10 feet diameter. The grate surface of the main boilers is 863 square feet and the heating surface 26,936 square feet; the heating surface of the auxiliary boiler is 900 square feet.

It was expected that 13,000 I.H.P. would be made with an air pressure of half an inch, giving a steady speed of 20 knots, and 20,000 I.H.P. under forced draft for occasional spurts of a few hours to give 22 knots. Owing to the failure of the *Vulcan* and other vessels, however, it was decided before the *Blake's* trials took place not to attempt a forced-draft run, but to be satisfied with the performance under assisted draft, if up to the estimate.

The trial occurred November 18, 1891, and lasted seven hours. The draught was 24 feet 8 inches forward and 26 feet 8 inches aft, with the designed displacement of 9,000 tons. The steam pressure at the boilers was 125½ lbs., the air pressure in fire

rooms, 0.42 inches, and the average revolutions, 88.41 starboard and 89.39 port. The other data are appended:

	Starboard.		Port.	
	Forward.	Aft.	Forward.	Aft.
Vacuum in condensers	27.85	27.85	28.1	28.1
Mean pressures in cylinders, H.P.	43.04	39.85	42.36	42.45
Mean pressures in cylinders, I.P.	31.49	30.82	30.17	28.38
Mean pressures in cylinders, L.P.	11.68	12.40	12.85	12.32
I.H.P., each engine	3,631.42	3,589.07	3,721.37	3,583.50
I.H.P., each shaft	7,220.49		7,304.88	
Aggregate	14,525.37			

During the trial four runs were made over the measured mile, and the mean speed was 19.12 knots. The mean speed by patent log for the seven hours was 19.28 knots.

The boilers gave no trouble under the easy steaming to which they were subjected, but the speed was more than three-quarters of a knot short. The disappointment has been very great, and there were rumors that the boilers would be replaced by others similar to those of the *Edgar*, but, as orders have been given to commission her for service, this has evidently been abandoned for the present.

Edgar.—This vessel has been described on p. 134 of vol. III. The progressive trials were completed on November 27th, 1891, with the results which will be given presently. The official contract trials took place shortly before; meanwhile the following additional detailed information in regard to the machinery will be of interest. It is abstracted from "Industries":

The engines are triple-expansion, twin-screw and vertical, with cylinder diameters of 40, 59 and 88 inches, and a stroke of 51 inches. The liners of H.P. and I.P. cylinders are of steel, and those of the L.P. of cast iron. The covers to cylinders and valve chests are of cast steel, as are the pistons. The piston rings for H.P. and I.P. are of phosphor bronze, and for the L.P. cast iron. No springs are fitted to the H.P. rings, which are one inch wide and separated by a distance piece; the other rings are $4\frac{3}{4}$ inches

wide. The H.P. is fitted with piston valves, but the other cylinders have double-ported flat slide valves.

"The piston type is now entirely discarded for the low-pressure cylinders, owing to the great difficulty of keeping them tight. This defect is not of so much consequence in the high pressure, since the steam leaking through can still do work in the intermediate and low pressure, and may also, being superheated by wire drawing, have some effect in reducing the initial condensation in the intermediate; but steam leaking past the low-pressure piston causes, of course, simple waste and worse."

The bearing surface of each crosshead slipper is 30 inches by 23¼ inches. The crosshead pins are of steel, 9 inches diameter and 15 inches long.

The crank shaft is in three sections and hollow. The crank pins are 16¼ inches diameter, with an 8-inch hole, and are 18 inches long. The shaft journals are 15¾ inches diameter, with a 7¼-inch hole, and each of the six is 18½ inches long. The propeller shaft is 15 inches diameter. The thrust is of the horse-shoe type, faced with white metal, and the rings have a diameter of 25 inches, the shaft being 15¾ inches. The propellers are of the Griffith type, three-bladed, and made of gun-metal. The diameter is 16 feet 9 inches; pitch, adjustable from 21 feet 10 inches to 24 feet 10 inches; it was set for the moderate speeds at 23 feet 4 inches, but altered for full speed to 24 feet 4 inches. The developed area of one screw is 54 square feet, and the projected area 39¾ square feet. The diameter of boss is 4 feet 6 inches.

The main condensers are of gun metal, the water passing through the tubes, and two independent centrifugal pumps are fitted to each. Each condenser has 4,300 tubes, ¾ inch in diameter and 8 feet long, giving a cooling surface of 6,751 square feet for each condenser, or 13,502 for both. This is 1⅛ square feet per I.H.P. (as designed) at full power, and, though much less than has been usual, appears to have given a good vacuum. The air pumps are 34 inches diameter and 21 inches stroke. All the valves are plain discs of sheet brass, ⅛-inch

thick, 6¼ inches diameter, and there are twelve in each set of foot, bucket and delivery valves.

There are four double-ended main boilers, with four Fox's corrugated flues at each end; diameter of shell, 16 feet; length, 18 feet. Each boiler has four combustion chambers, each pair of furnaces at each end having one in common. The depth of each chamber is 23 inches. The furnaces are 3 feet 6 inches in diameter, and the grates are 7 feet long. There are 1,176 tubes, 2½ inches outside diameter, and 6 feet 7¾ inches between tube plates, giving each boiler a tube surface of 5,009 square feet. The heating surface of furnaces and combustion chambers is respectively 346 and 364 square feet, and of the tube plates 201 square feet, making the aggregate for each boiler 5,920 square feet. The grate surface is 203 square feet. Thus for the nominal I.H.P. at full power of 3,000, there are $\frac{1}{1.5}$ square foot of grate surface, 1.67 square feet of tube surface, and 1.973 square feet of total heating surface. The working pressure is 155 pounds. Each boiler has three 5⅝-inch safety valves. The main steam pipe is 14 inches diameter, and wound with copper wire. There is one auxiliary boiler with 56.25 square feet of grate surface in three furnaces, each leading to a separate combustion chamber. This boiler has supplied 840 I.H.P. for the auxiliaries.

There are two auxiliary condensers, two Weir's evaporators and two Normandy distillers, besides some fifty-two auxiliary engines, including pumps.

The contract requirements were 10,000 I.H.P., with not over ½-inch air pressure—the so-called natural-draft trial—and 12,000 I.H.P., with not over 1 inch air pressure, the maximum revolutions to be 100 per minute. Under an air pressure of 0.3 inch the I.H.P. of the various cylinders was—starboard, H.P., 1,388; I.P., 1,698; L.P., 1,986; total, 5,072; port, H.P., 1,356; I.P., 1,595; L P., 2,156; total, 5,107; thus giving an aggregate I.H.P. of 10,179. The revolutions were 99.2 and the mean speed 19.25 knots. The pitch of the propeller was 23 feet 4 inches, so that the slip was 15.8 per cent. As the revolutions had nearly reached

the allowable maximum, it was decided to increase the pitch for the full-power trial to 24 feet 4 inches.

With an air pressure of a trifle less than ¾ inch the I.H.P. was:

	Starboard.	Port.
H.P.,	1,730	1,910
I.P.,	2,031	1,844
L.P.,	2,395	2,553
Total,	6,156	6,307

giving an aggregate of 12,463, with a speed of 20.97 knots, 104.5 revolutions, and 16.4 per cent. slip.

During the progressive trials in Stokes Bay, the maximum I.H.P. was 13,101. The speed, however, was only 20.49 knots. This was partly due to the "drag" consequent upon the comparatively shallow water, and partly to the draught being somewhat greater—6½ inches.*

These trials gave great satisfaction in England after the failures of the *Blake* and the *Vulcan*, and it appears to have been due entirely to the better design of the boilers. We give under "Notes" an article treating of this question.

SPAIN.

The new navy which Spain is rapidly building is to be organized in three divisions, with headquarters at Cadiz, Ferrol and Cartegena. The Cadiz division will consist of the *Emperador Carlos V.*, a ship similar to the *Blake* of the English Navy; the two armored cruisers, *Infanta Maria Teresa* and *Princesa de Asturias*, each of 7,000 tons; the protected cruiser, *Reina Regente*, of 5,000 tons; two third-class cruisers, two torpedo gun vessels, and three torpedo boats. The Ferrol division will consist of a sister ship to the *Carlos V.;* the 7,000-ton cruisers, *Quendo* and *Cardenal Cisneros;* the protected cruiser *Alfonso XIII.*, of 4,800 tons; two third-class cruisers, two torpedo gun vessels, and three torpedo boats. The Cartegena division will consist of the battleship *Pelayo;* the 7,000-ton cruisers *Viscaya* and *Cataluna;*

* During these trials a speed of 10 knots was obtained for 847 I.H.P., and 16 knots for 2,000 I.H.P.

the protected cruiser *Lepanto* of 4,800 tons; two third-class cruisers, two torpedo gun vessels, and three torpedo boats. The old ironclads *Victoria* and *Numancia* are to be re-engined and re-armed. The cruiser *Reina Cristina* is to be fitted as a transport, the cruiser *Reina Mercedes* as a cavalry transport and storeship, and the cruiser *Alfonso XII.*, as a magazine and torpedo-depot ship.

GERMANY.

Kaiserin Augusta.—The *Kaiserin Augusta* is a recent addition to the German Navy, and the following data, although very incomplete, comprises all the knowledge at present obtainable.

Her displacement is 6,026 tons. The collective I.H.P. for which the engines were designed is 12,000, and the three screws with which the vessel is fitted are intended to develop a speed of 20 knots per hour.

For ordinary cruising, at a speed of 12 knots, the middle screw alone will be used. The two outside screws, without the middle one, are expected to give a speed of 18 knots.

FRANCE.

Torpedo-Depôt Ship.—France, which has not hitherto possessed a torpedo-depôt ship, is about to construct one after designs which have been prepared by M. J. C. Duplaa-Lahitte, of the Corps du Génie's Maritime. The vessel, which is to be named the *Foudre*, will be 370 feet 6 inches long and 51 feet 3 inches broad, and at a mean draught of 20 feet, will displace 5,970 tons. Engines of 11,400 aggregate H.P. will drive twin-screws and give an extreme speed of 19 knots. The armament will consist of eight 3.9-inch, four 2.5-inch, and four 1.8-inch quick-firing Canet guns, and five torpedo-ejectors; and she will carry ten torpedo boats of the second class, which will be hoisted and lowered by means of hydraulic gear. A certain amount of protection will be given to the vessel by a steel deck 1½ inches thick. The *Foudre*, which will be ready for sea in 1895, will be supplied with material and apparatus for the repair of torpedoes and torpedo boats and for the construction of small craft.

Eclair, *Orage* and *Kabyle*, the new torpedo boats for the French Navy.—The final speed trials have just been made of the *Orage*, the second of the three first-class torpedo boats—the *Eclair*, *Orage* and *Kabyle*—lately completed for the French Navy by the Société des Forges et Chantiers de la Mediterrannée at their La Seyne shipyard, with very satisfactory results, a speed of 21.6 knots per hour, having been obtained under forced draft on the measured distance, being an excess of 1.6 knots on the specified speed of 20 knots. During the trial the whole of the machinery worked very satisfactorily.

The first completed of the three sister vessels, the *Eclair*, gave results almost as good on her speed trials made during the month of October last, obtaining a maximum speed of 21.46 knots, maintained for one hour, after running for the previous five hours at a mean speed of 18 knots per hour. During these trials the pressure of the steam in the boilers was only 185 lbs. per square inch, in place of 200 lbs. allowed by the contract, and with an air pressure in the stokehold of only 1.18 inches.

On the six-hours trial conducted to determine the consumption of coal by the boilers of the *Eclair* at a speed of 10 knots per hour, it was found that 9.6 lbs. were required per square foot of grate surface, being 14.7 lbs. per mile run. As the *Eclair* carries 14½ tons of coal, she is enabled, according to this result, to steam a distance of about 2,165 knots at a speed of 10 knots in place of 1,650 set forth in the specification.

The *Eclair*, *Orage*, and *Kabyle* are of the *Agile* type, and like this boat and her sister, the *Active*, have been designed by Monsieur Lagane, the chief of the Société's shipyard at La Seyne, near Toulon.

They have a length of 143 feet, a beam of 14 feet 9 inches, an extreme draught of 8 feet 3 inches, and a displacement of about 104 tons. The material of construction is mild steel throughout, the whole of the shell plates, frame angles, and bulkhead plates having been galvanized before being riveted together.

Incandescent electric lamps are used for the interior lighting of each of the three boats, and a small electric search-light projector is fitted just forward of the forward funnel. There are

two torpedo firing tubes, one situated forward and one aft. One torpedo is stowed in each of these two tubes, and there are in addition two stowed in reserve, making four in all. The armament is completed by two Hotchkiss 37-millimeter quick-firing guns, one mounted on each side of the deck.

The engines, in which brass has largely been used as the material of construction, even the bed plates being brass castings, were constructed by the shipbuilding company at their engine works in Marseilles. They are built to develop 1,200 I.H.P. at the maximum speed, and are supplied with steam by two Thornycroft boilers, working at a pressure of 200 lbs. per square inch. These three boats will shortly enter into commission, and form a valuable addition to the French Mediterranean mosquito fleet.

CHILE.

The *Capitan Prat* is a sea-going armored cruiser built at La Seyne, France, for the Republic of Chile. The following data are compiled from various authorities:

Displacement,	6,900 tons.
Length,	328 feet.
Beam,	60 feet 8 inches.
Maximum draught,	21 feet 10 inches.
Cost,	£391,000.
Side armor belt (thickness),	12 inches.
Side armor belt (length),	complete.
Battery,	4 9.45 inches, 8 4.72 inches B.L.R., 4 6-pdr. R. F., 6 R. C.
Speed,	19 knots.
Horse-power,	12,000.
Coal,	400 tons.

The motive power for working her guns is electricity, but provision is made for operating the battery by hand in case of injury or non-operation of the electric gear.

The motive machinery is in two water-tight compartments, and is protected by armor of 6.5 inches thick.

The ship is propelled by twin screws.

The steel hull is sheathed with wood, and coppered.

RUSSIA.

Georgei Pabeidonosets (*George the Victorious*).—This vessel has been under construction at Sebastopol by the Russian Steamship and Commercial Company since 1889; is now ready for sea.

The following data are quoted from "The Marine Engineer" of Feb. 1, 1892:

Length,	339.5	feet.
Beam,	69	feet.
Draught,	26.58	feet.
Displacement at this draught,	10,280	tons.

The machinery and boilers were made by Messrs. Maudslay, Sons & Field, and under forced draft the I.H.P. to be developed is 16,000, giving a speed of 17.5 knots per hour. She is provided with twin screws.

The new vessel, will be armed with six long-range 12-inch guns, mounted *en barbette*, seven 6-inch guns mounted on the battery deck, eight quick-firing guns of the Baranovsky model, six 37-millimeter quick-firing guns for the tops, and seven torpedo propelling tubes.

MERCHANT STEAMERS.

FREIGHT STEAMERS FOR THE MINNESOTA STEAMSHIP COMPANY.

Contributed by Mr. ERNEST N. JANSON, Associate.

Among some of the latest contracts given out to lake shipbuilders are two large steamers for the Minnesota Steamship Company of Cleveland, Ohio. They are both intended for use in the Lake Superior ore trade, and will be the largest vessels on the lakes. One is being built at the Globe Iron Works Company, of Cleveland, Ohio, the other at the Chicago Shipbuilding Company, Chicago. The ships are duplicates in all respects, being constructed upon the same lines; the engines and boilers are being built by the Globe Iron Works Company, and are alike for both ships. The hulls are steel throughout, and, like all modern lake freight steamers, will have a double bottom

extending the entire length from the stuffing-box bulkhead to the collision bulkhead. Dimensions are as follows: Length over all, 348 feet; breadth (moulded), 45 feet; depth (moulded), 24 feet 6 inches. The inner bottom is at the center longitudinal plate 4 feet 6 inches deep, and divided in eight water-tight compartments. Solid floor plates 20 inches deep run from the center plate to the bilges and connect with brace plates and futtocks.

On each side of the center longitudinal plate are three longitudinal girders, on top of which the Z-bars rest to take the inner bottom plating. Belt frames are built in at every 10 and 12 feet, side keelsons and stringer plates as usual. As a consequence of great carrying capacity on a small draught the fullness of model is a striking feature, the block coefficient being .802 and displacement 6,000 tons at 16 feet draught. The bottom is nearly flat with only a 3-inch rise of floor, the bilges turn with a short curve and the sides with a 6-inch "tumble home" at the rail. There are two decks of which the upper one is steel uncovered; bulwarks of wood, and one plain mast, placed forward, constitutes the rigging. On the upper deck are nine hatches 32 feet wide by 8 feet long; all houses, with arrangements of staterooms for officers and crew, navigation house, skylights, etc., are located on the spar deck.

One set of triple-expansion direct-acting jet-condensing engines actuates each screw. The screws have adjustable blades. The cylinders are without liners or jackets and of the following diameters: 24 inches, 39 inches, 63 inches, with a common stroke of 48 inches. Connecting rod 12 feet between centers. One single-ported piston valve on H.P., a slide single-ported valve on I.P. and a double-ported slide valve on L.P. The crank shaft is built up and in three interchangeable sections, the eccentrics being placed on the couplings. All journals are lined with magnolia metal in strips. All valves are operated by the Stephenson double-bar link motion, reversing by a combined steam and hydraulic reversing engine operated from the working platform. Adjustable blocks for cutting off are fitted on all suspension arms. The thrust bearing is of the ordinary horse-shoe type with movable yokes. The air pump is driven from the I.P. cyl-

inder crosshead, and with this pump, and by one crosshead, are worked two bilge pumps, one fire pump, and one feed pump. For emptying the water-bottom tanks, which will hold about 1,400 tons of water, are attached one centrifugal pump and one duplex pump, each with 9-inch suction. There is also an independent duplex pump for feeding boilers. Besides these auxiliaries there is a Providence steam windlass, two steam capstans, steam steerer and a double-cylinder deck hoist, both of Globe make. Three Scotch boilers, side by side, are located on the main deck in a recess 3 feet below the deck. Their dimensions are 12 feet diameter by 12 feet long, and have three Adamson's furnaces 38 inches diameter in each boiler. Steam pressure 175 pounds absolute. Heating surface 4,562 square feet, and 183 square feet of grate.

Main shafts and connecting rods are made of wrought iron, piston rods, eccentric rods and gear of machinery is of steel. All steam and exhaust pipes of lap-jointed steel plates; feed pipes drainage pipes, etc., of wrought iron. Coal bunkers will hold about 200 tons, and the total carrying capacity of the vessel at a draft of 16 feet is 4,000 gross tons. Calculated speed 14 miles an hour. A complete electric light plant is to be installed in each ship.

Almy.—Through the courtesy of Mr. T. Jackson Shaw, superintending engineer of the Harlan & Hollingsworth Co., of Wilmington, Del., we are enabled to give the results of the progressive trials of this yacht and some of the data of hull and machinery.

Length between perpendiculars, 155 feet 6 inches; length on deck, 180 feet; beam, 24 feet; depth, 14 feet; draught, on measured mile trials, 8 feet 5 inches forward, 10 feet aft, 9 feet $2\frac{1}{2}$ inches mean; displacement during trials, 397.3 tons; immersed midship section, 160.6 square feet. Draught on coal-consumption trials, forward 9 feet $6\frac{3}{4}$ inches, aft 10 feet 5 inches, mean 9 feet $11\frac{7}{8}$ inches; displacement, 455.4 tons; immersed midship section, 179.6 square feet.

There is a single-screw triple-expansion engine with cylinders 17, 28 and 42 inches diameter by 22 inches stroke. The con-

denser which forms part of the engine framing has 1,522 square feet of cooling surface. The air pump is worked by levers from main engine, and is 20 inches diameter by 8 inches stroke. The circulating pump is centrifugal and driven by an independent cylinder, 6 inches diameter and 8 inches stroke; the diameter of outlet and inlet of pump is 8 inches.

The H.P. cylinder is fitted with a piston valve, the intermediate and low-pressure cylinders have double-ported slide valves.

All valves are worked by a link motion, and each valve is independent of the others.

The links are of the usual bar type.

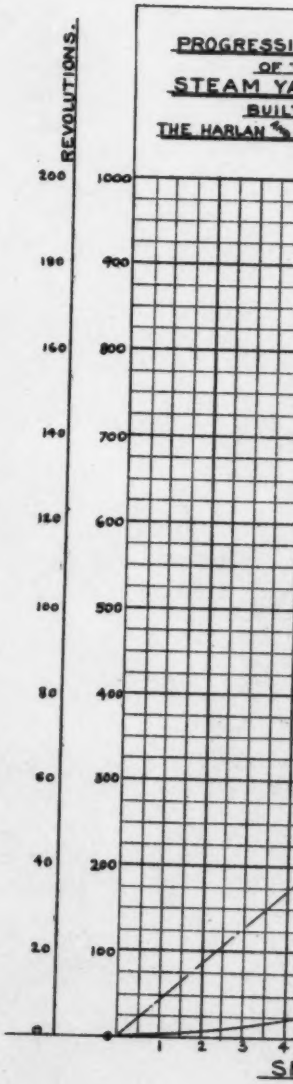
The shafting and piston and connecting rods are of wrought iron; diameter of crank shaft and pins, 7½ inches; of line shaft, 7¼ inches; of propeller shaft, 7½ inches.

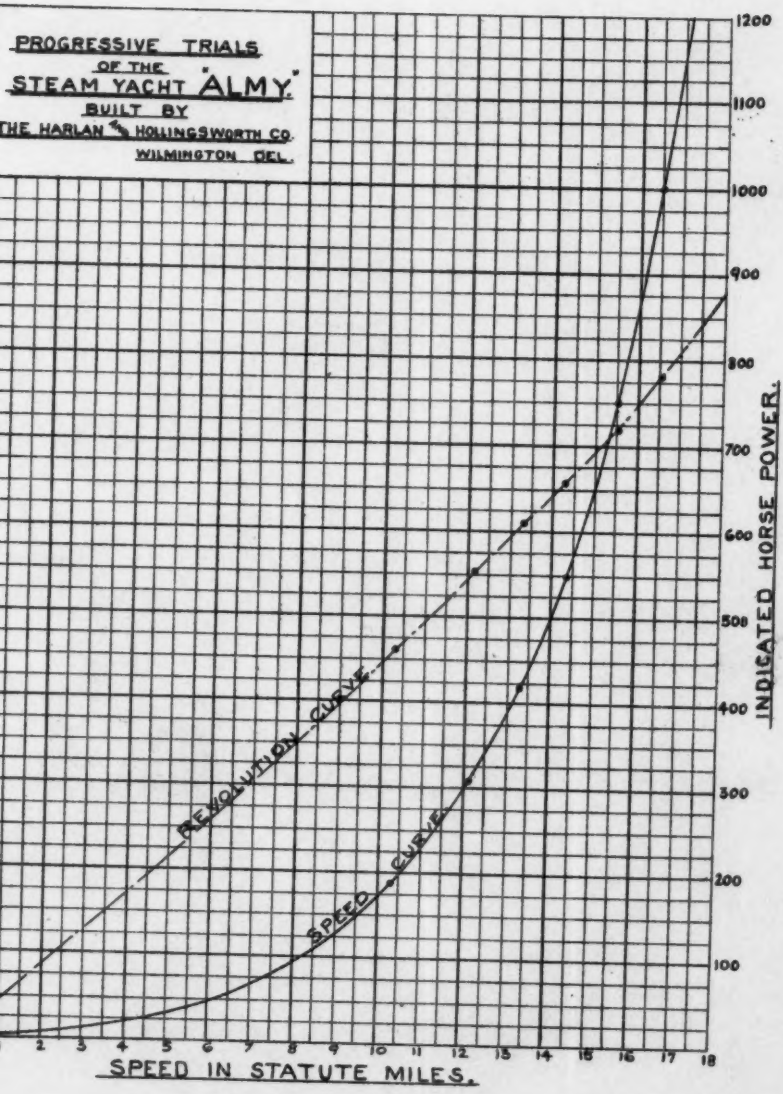
The propeller is a true screw of the built-up type with four blades; diameter, 8 feet; pitch, 11 feet; helicoidal area, 21 square feet.

The boiler is of the ordinary cylindrical type, 12 feet 6 inches diameter, and 12 feet long inside heads, with 3 corrugated furnaces of 42 inches external diameter; working pressure, 170 lbs. There are 302 tubes, 2½ inches external diameter and 8 feet 9 inches long. The grate surface is 75 square feet and the heating surface 2,067 square feet. The grate bars were intended to be 6 feet 6 inches long, but were made 7 feet 6 inches at request of engineer of the vessel.

Forced draft on the closed fire-room plan is used for full power, a Sturtevant blower with a capacity of 7,500 cubic feet per minute being fitted. The feed pump is a Worthington duplex, Admiralty pattern, with 5¼-inch steam and 3½-inch water cylinders, and 5-inch stroke controlled by a float in feed tank.

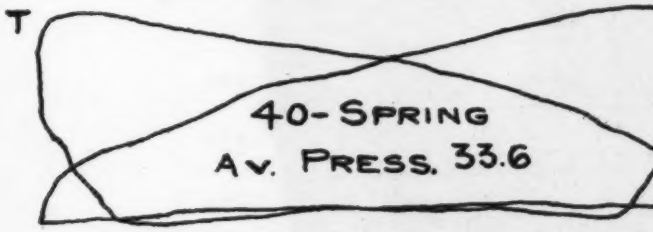
The progressive trials were made over a carefully measured course in Delaware Bay, one statute mile in length, range poles being set up on shore of a size readily distinguished. There was one run each way for each spot on the curve, care being taken to keep the revolutions uniform during each run. No record was kept of the air pressure under forced draft, but the highest observed was about 1½ inches of water. The results are given in the following table:


PROGRESSI
OF
STEAM YA
BUIL
THE HARLAN
REVOLUTIONS.
200
190
160
140
120
100
80
60
40
20
0
1000
900
800
700
600
500
400
300
200
100
1
2
3
4
SI

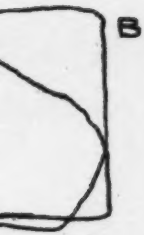

PROGRESSIVE TRIALS
OF THE
STEAM YACHT "ALMY"
BUILT BY
THE HARLAN & HOLLINGSWORTH CO.
WILMINGTON DEL.
REVOLUTION CURVE
SPEED CURVE
INDICATED HORSE POWER.
SPEED IN STATUTE MILES.
100
200
300
400
500
600
700
800
900
1000
1100
1200
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18

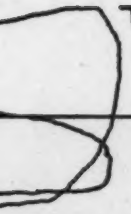
H. P. : 346 I.H.P.

I. P. 363.7 I.H.P.



L. P. 314. I.H.P.





STEAM YACHT "ALMY".

ENGINE - $\frac{17"-28"-42"}{22"}$

JULY 9TH 1891.

BOILER PRESS	162
1ST RECEIVER	59
2ND "	12
VACUUM (INCHES)	22½
REVOLUTIONS	158

THROTTLE WIDE OPEN.

I.H.P.	346 -- H.P.
"	363.7 -- I.P.
"	314 --- L.P.
TOTAL	1023.7

Number of run	1	2	3	4	5	6	7	8	9	10	11	12
	up.	down.	up.	down.	up.	down.	up.	down.	up.	down.	up.	down.
Steam pressure in boiler	115	130	150	151	158	160	157	140	152	158	160	162
Steam pressure in 1st receiver (per gauge)	5	6	14	15	22	23	29	34	42	43	57	59
Steam pressure in 2d receiver (absolute)	7.5	8.5	9.5	10	12	13.5	0.5	1.5	5	5	10	12
Vacuum in inches	26	26	26	26	25.5	25.7	25.7	25.5	24.7	24	23	22 5
Revolutions of main engine per minute	91	96	110	112	121	123	127.5	132 5	142	146	156	158
Throttle	D.	D.	D.	D.	D.	D.	D.	D.	D.	D.	O.	O.
	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.	m. s.
Time on mile	5-31	6-7	4-24	5-36	3-58	5-12	3-42	4-55	3-22	4-29	3-19	4-04
Mean speed in statute miles	10.342		12.175		13.332		14.21		15.6		16.425	
Slip in per centum	11.68		13.6		14.18		14.45		15.4		17.1	
Mean I.H.P.	185.0		306.7		421.5		550.0		747.0		999.2	

D Throttled down to secure desired revolutions. O Throttle open.

There was a light wind in the direction of course and the weather was fair.

The progressive trials were made under Mr. Shaw's personal direction. The coal-consumption trials were made subsequently by the engineer of the vessel, but, unfortunately, the horse-power was not taken. Each trial lasted five hours, and gave a result of 435 pounds of coal per hour for a speed of 11 statute miles and 966 for 13.6 statute miles. The coal was Lehigh anthracite, egg size, burned with natural draft.

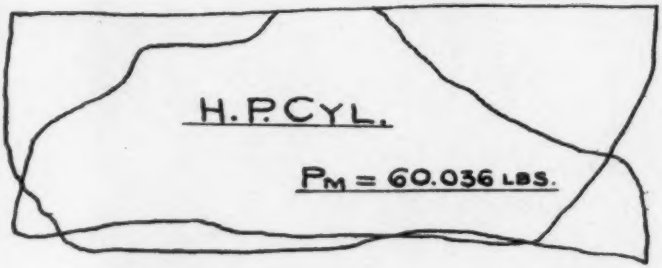
La Touraine.—By the courtesy of Mr. C. G. West, Superintendent, and through the kindness of Mr. Thos. H. Barrett, U. S. Local Inspector of Steam Vessels, the JOURNAL presents indicator cards, on the opposite page, from the engines of the S. S. *La Touraine*, of La Compagnie Générale Transatlantique. Diameter of cylinders, 40.945, 60.63, and 100.00 inches by 65.355 inches stroke. Boiler pressure, 135; cut-off, .71; vacuum, 26.77 inches.

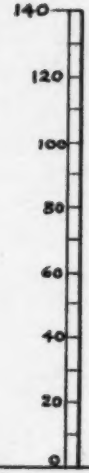
A full description of this vessel may be found on page 432, Vol. II, and page 611, Vol. III, of the JOURNAL.

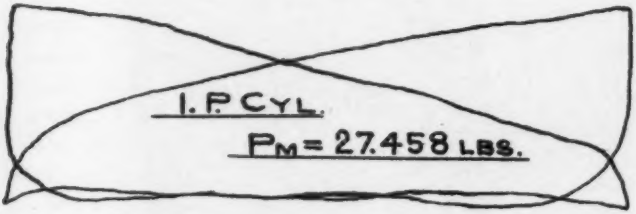
S. S. Tekoa.—The recent performance of the *Tekoa*, built and engined by Messrs. W. Gray & Co., of West Hartlepool, for the New Zealand Shipping Co., is a very striking example of the efficiency of the modern cargo boat, and we give her as an example of the point to which careful design and workmanship have brought the camels of the sea. This vessel is of 2,646 tons register, 4,050 tons gross, and has a deadweight capacity of 6,250 tons. Her refrigerating chambers are capable of carrying the carcases of 70,000 sheep. The engines are of the triple-expansion type, with cylinders 27 inches, 43 inches and 72 inches diameter, and 45-inch stroke. Steam at 160 pounds pressure is supplied by three large double-ended boilers, with twelve furnaces in all. Her speed on trial was 11½ knots. On her outward voyage she ran from Teneriffe to Auckland, 12,059 knots, without a stoppage and without slackening speed. On this trip the actual steaming time from London to Auckland was 58 days 2 hours, and the average speed was 10 knots, with a consumption of 21 tons 5 cwt. per diem for all purposes, including the work-

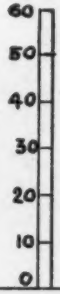
S.S. "LA TOU

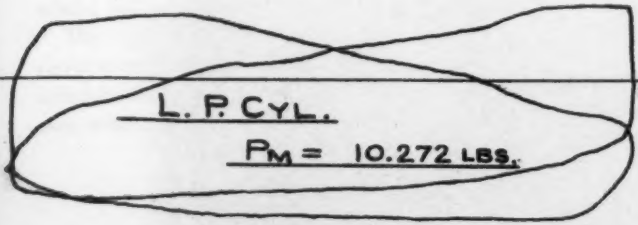
PORT ENGINE.

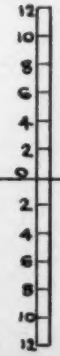




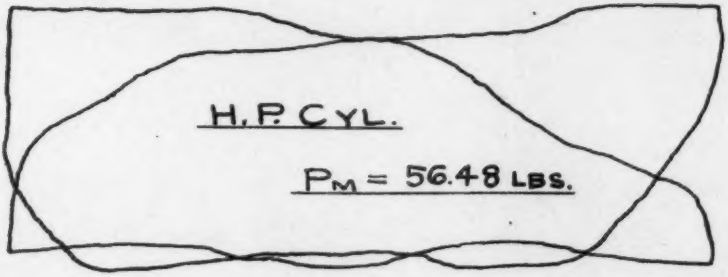


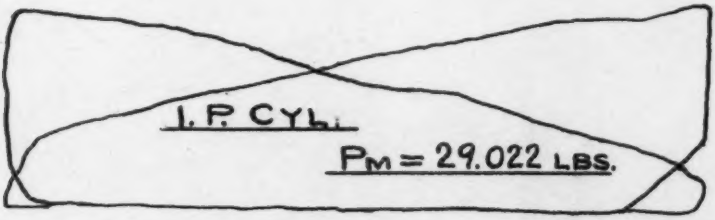


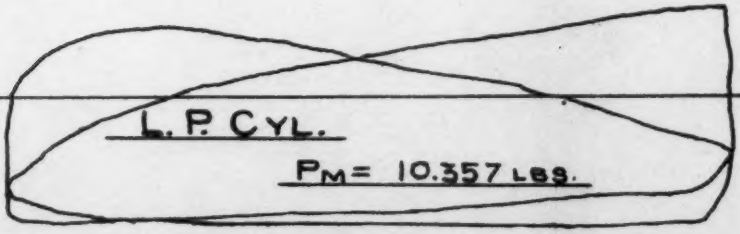




STARBOARD ENGINE.



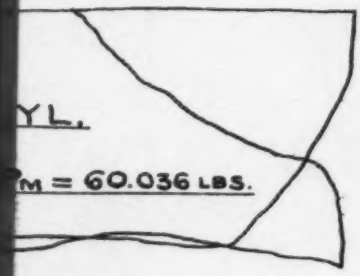


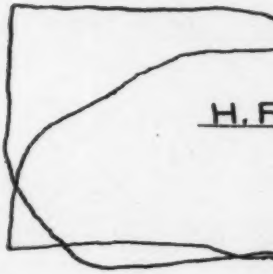


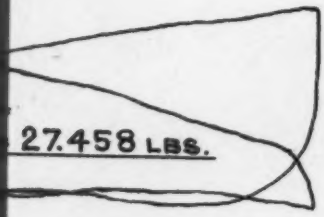
S.S. "LA TOURAINE".

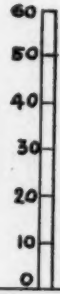
NGINE.

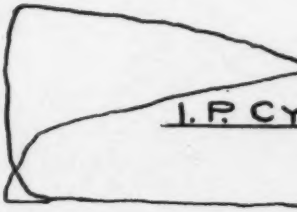
STARBO

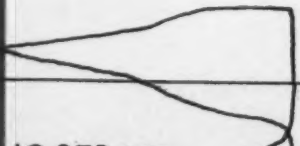


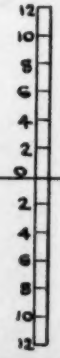


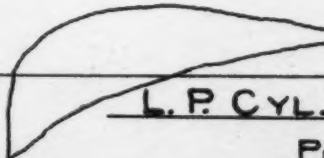




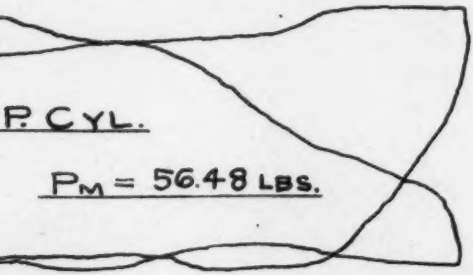








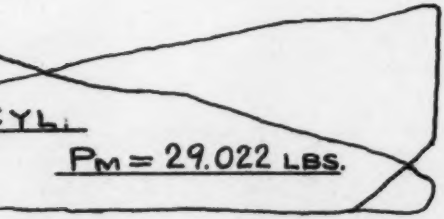
OARD ENGINE.



P. CYL.

P_M = 56.48 LBS.

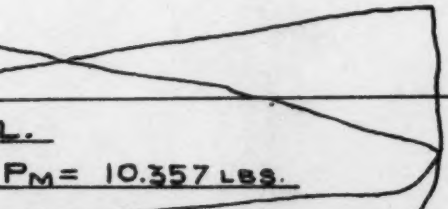
	PORT.	ST'B'D.
H. P.	2061.4	1939.2
I. P.	2066.2	2185.
L. P.	2103.6	2072.8
	6231.2	6197.0



CYL.

P_M = 29.022 LBS.

TOTAL I.H.P. 12428.2



L.

P_M = 10.357 LBS.

ing of the auxiliary engines. Now we find all the regular mail steamships to New Zealand year after year doing two and a half circuits of the globe, a distance of say 66,000 miles in the twelve months, and Atlantic vessels occasionally completing thirteen round voyages, which amounts roughly to 78,000 nautical miles, in the same period, whilst these New Zealand cargo vessels run the full distance from England to New Zealand, upwards of 13,000 miles, without a check, and no doubt could go further if any object were to be attained by doing so. The *Tekoa* carried her 6,250 tons of cargo at a speed of 10 knots, on an expenditure of about 1,237 tons of coal, or just under 4 cwt. of coal per ton of cargo. To carry the analysis of this performance one step further, she transported a ton of cargo one mile by consuming half an ounce of coal. The *Tekoa*, with her boiler pressure of 160 pounds, burned something like 1.65 pounds of coal per indicated horse-power per hour. Higher pressures for cargo boats seem already in the field; we hear of one just ordered by the West India and Pacific Line, with a boiler pressure of 200 pounds. This vessel is being built by the Naval Construction and Armaments Co., and as the difficulties which are at present found in the use of higher pressures are gradually resolved, so will the consumption for indicated horse-power be decreased, and the expense of running a steamer will fall in proportion.—*Marine Engineer* (London).

The Royal Mail Steamer *Ophir*.—The *Ophir*, which is built of mild steel, was launched from the yard of Messrs. Robert Napier & Sons, on the Clyde, on the 11th April, 1891. She has a straight stem and an elliptical stern, her length over all being 482 feet, whilst she is 465 feet between perpendiculars. The extreme breadth is 53 feet 6 inches. The moulded depth is 37 feet. She draws 24 feet 6 inches at load line, and her displacement is thus 10,550 tons. Her gross registered tonnage is 6,910 tons, whilst the net is 3,222.79 tons.

The vessel is built to class 100 A1 at Lloyd's, and fulfills the requirements of the Admiralty for armed cruisers. The frames amidships are 27 inches apart. The shell plates are of medium size, with ordinary straps below the water line and double straps

in the upper strakes. For the sake of lessening the rolling deep bilge keels are fitted on either side. The hull is coated with Hartmann's Rahtjen's composition. There is a longitudinal bulkhead in the engine room, and the two boiler rooms are separated by a coal bunker 30 feet long, so that the chance of a blow striking the vessel and filling both boiler rooms is much reduced. Besides this there are twelve transverse bulkheads, all carried to the upper deck. Where doors have been necessary, they are made to slide vertically in closing, and there is a tell-tale on the upper deck, whereby it can be immediately seen whether the door is open or shut, and from this position the doors can be worked. The double bottom extends nearly the whole length of the vessel. It has a capacity of 900 tons, and the space between the inner plating and the skin of the ship is 3 feet 9 inches.

The cargo gear consists of a derrick at the fore hatch fitted with swinging and topping rams, and capable of working either side of the ship. This will raise a ton. A small hoist is also provided of half this power for bringing up baggage wanted on the voyage. At the main and after hatches there are four deck cranes. These will raise 45 cwt. 60 feet or $4\frac{1}{2}$ tons 30 feet, and swing it over the vessel's side. The water used in the whole system is supplied at 1,000 lbs. pressure by one of Messrs. Brown Brothers' own engines, which is built on a steam accumulator. To avoid any chance of a breakdown, however, a small independent pump is also supplied. This is capable of supplying enough power to drive the steering gear and to raise baggage, and thus any repairs to the main pumping engines can be effected at sea, and the discharge of cargo in port is thus not delayed whilst the appliances are put in order.

Like all recent ocean passenger steamships, the *Ophir* is lighted throughout by electricity. There are upwards of 850 lamps of 16-candle power, and there is also a complete plant for use in the Suez Canal. This consists of a 24-inch projector with cage, and 25-ampere mast-head arc lamp. The ship is divided into twenty circuits, and on some of these the lamps are grouped, for places to which the light of day cannot penetrate, and are continu-

ously alight. The power is supplied at a pressure of 65 volts by four compound slow-speed engines, and fitted with governors capable of adjustment by hand whilst the engines are running. The cylinders are 8 inches and 13½ inches diameter, with a 7-inch stroke. The speed is only 230 revolutions. The dynamos are direct-compound, wound with drum-wound bar armatures. The apparatus is very compact, the engine and dynamo only occupying a floor space of 8 feet 11 inches by 2 feet 11 inches. One dynamo is capable of supplying 325 lamps, and accordingly three have more than sufficient power to light the entire ship, thus one is always kept in reserve. Any one of the dynamos can be used for the projector. They are always run in parallel for circuit feeding.

The main engines are separated from each other by a longitudinal bulkhead, and consist of two sets of triple-expansion engines having H.P. cylinders 34 inches, intermediate 51½ inches, and low-pressure 85 inches in diameter, the stroke being 54 inches. The cylinders are all steam-jacketed. Piston valves are fitted to the high-pressure and intermediate cylinders, and a double-ported slide valve to the low pressure. The working parts are all made of steel. The engines are fitted with Dunlop's patent combined steam and pneumatic marine governors, which have been successfully fitted to some of the most powerful vessels afloat. This governor consists of a small air vessel placed as far aft in the ship as possible. At its lower end it is connected to a sea cock, so that sea water can be admitted. A small pipe is carried from the top of this vessel through the shaft tunnel to the under side of an apparatus fitted in the engine room. The pipe communicates with an elastic rubber diaphragm at the bottom of the apparatus, and a spindle with a balance spring is placed above it. When the air vessel and piping are full of air, the sea cock is opened, the air contained is compressed to a density proportionate to the depth below the surface of the water in which the propeller is revolving. The spring is then screwed up to balance the pressure of this head of water, and the apparatus consequently remains stationary as long as the screw is immersed at that depth, but as

soon as the vessel pitches and the stern goes up, the head of water at the stern above the air vessel decreases, and the air pressure consequently decreases also. The spring at once expands, and opens the steam passage to a little governor cylinder. This closes the steam valve of the main engine to an extent proportionate to the amount which the stern rises. Thus for a slight pitch the engines are but slightly checked, whilst in a heavy plunge steam is entirely shut off.

Each engine room is fitted with a complete and independent set of pump and auxiliary engines. There is one of Messrs. G. & J. Weir's feed heaters, which is capable of heating all the feed water to 212° Fahrenheit. The firm supply no less than four of their direct-acting feed pumps which work independently of the main engines. One pump takes the water from the tank and passes it into the feed heater, whilst another draws it from the feed heater and supplies it to the boilers. The diameter of the cylinders of these pumps is 16½ inches, and of the bucket 12½ inches, whilst the stroke is 30 inches. The speed of the pumps is regulated by automatic-controlling gear on the fresh-water tank, and also on the feed heater.

There are four centrifugal circulating pumps—two in each engine room—working independently of the main engines, for pumping the water through the condensers. One of these pumps is equal to the duty required in each room. There are two sets of air and bilge-pumps to each engine. These are worked off the crossheads of the high and low pressure engines. The sanitary pumps are also worked off the main engines.

There are donkey pumps of the Worthington type for working to the bilges, and ballast tanks, washing decks, &c. The ashes are raised from the stokeholds by means of hydraulic hoists, so that as little noise as possible is made by the operation.

In each engine room there is a Weir's evaporator. These are capable of producing 25 tons of fresh water each per day of 24 hours, and this is applicable either for making up the water for boiler feed or for culinary purposes.

There are seven main boilers in two compartments. These

are separated by a coal bunker 30 feet long, so that the possibility of any injury flooding both groups is very remote. The after compartment contains three double-ended boilers, whilst there are two double-ended and two single-ended boilers in the fore compartment. The boilers are fired fore and aft, the two single-ended boilers being set back to back. There are coal bunkers alongside the top of the boilers and engines, thus affording them protection about the water line. The capacity of the bunkers is equal to 15½ days' consumption at 16 knots. But if the holds were also filled with coal she could cruise for 130 days at 10 knots, or steam 14,000 miles at full speed.

The main boilers are all 15 feet 1 inch in diameter, whilst the length of the double-ended boilers is 19 feet 1 inch, and the weight empty is 73 tons 16 cwt. The single-ended are 10 feet 8 inches long, and weigh 41 tons 16 cwt. The heating surface is 26,004 square feet. There are 36 furnaces with Purves' ribbed flues, the grate area being 756 square feet. The steam pressure is 160 pounds. The boilers are to be worked with natural draft and open stokeholds, but with a view to Admiralty requirements the stokehold is made capable of being shut in, and eight fans are provided, two in each stokehold, each driven by a separate Gwynne engine. These are placed at the bottom of a downcast shaft.

Beside the main boilers there is a distilling boiler fitted into a recess in the fore end of the after boiler room.

The propellers are three-bladed, with manganese-bronze blades, the diameter being 17 feet and the pitch 23 feet 3 inches. They project outwards 11 feet 7¼ inches from the center line of the vessel. The shafting extends 39 feet outside the hull, and is supported by brackets of cast steel made in two pieces. The upper arms are hollow, and fitted with a gauge rod to give notice of any wearing of the bearing. These gauge rods were fitted to the shafts of the *City of Paris* after her accident. The *Ophir* went for her speed trials on the 22d of October, when her best performance was 18¾ knots with 103 revolutions. She left the Clyde on the 24th October for the Thames, and steamed from the Cloch Light to Southend in 48 hours, in course of which

she underwent her 500-mile trial. This showed a result of 16 knots, with a consumption of 110 tons per day, and a mean indicated H.P. of 7,200. She left London for her first voyage on the 6th November, the date advertised for her sailing months before. Strikes had occurred on the Clyde since the date was fixed, and very little time was thus allowed between her arrival in London and her starting on her maiden trip. Thus she has perhaps hardly done herself justice, but for the sake of comparison we append details of her maiden voyage, and also those of that of the *Ormuz*, the most speedy, on the whole, of her predecessors.

ORMUZ, 1887.	*OPHIR, 1891.*
Left Plymouth 12.30 p. m., 5th Feb.	9.30 p. m., 7th Nov.
Off Gibraltar 6 p. m., 8th Feb.	9 a. m. to 6 p. m., 11th Nov.
Arrived Naples 6 p. m., 11th Feb.	2 p. m., 15th Nov.
Left Naples 10 p. m., 12th Feb.	2 p. m., 17th Nov.
Arrived Port Said 8 a. m., 16th Feb.	11 p. m., 20th Nov.
Left Suez 11.30 p. m., 18th Feb.	5 a. m., 22d Nov.
Arrived Aden 2.30 p. m., 22d Feb.	
Left Aden 10.30 p. m., 22d Feb.	
Arrived Colombo	6 a. m., 2d Dec.
Left Colombo	11 a. m., 3d Dec.
Arrived Albany	10 a. m., 13th Dec.
Left Albany	2 a. m., 14th Dec.
Arrived Adelaide 14th March	5 a. m., 17th Dec.

We cannot make a detailed comparison between these two vessels throughout the entire voyage as they did not make calls at the same ports after leaving Suez, but we find that in the earlier stages their times were:

	ORMUZ.			*OPHIR.*		
	d.	h.	m.	d.	h.	m.
Plymouth to Gibraltar, . .	3	5	30	3	11	30
Gibraltar to Naples, . .	3	0	0	3	20	0
Naples to Port Said, . .	3	10	0	3	9	0

It will be noticed how the *Ophir* improved towards the end of the voyage.

Thus, over the entire voyage the *Ormuz* was 37 days, and the *Ophir* 40 days. But whilst the *Ormuz* only spent 3 days 3½ hours in ports or in going through the canal, the *Ophir* spent 5 days 16 hours in port.—*Marine Engineer* (London).

ANNUAL MEETING.

The *Annual Meeting* for the election of officers and the transaction of other business took place at the Navy Department, Washington, D. C., December 21, 1891.

Papers announced to be read were by Passed Assistant Engineer G. W. Baird, U. S. N., on *Electric Light Fittings*, and by Passed Assistant Engineer W. F. Worthington, U. S. N., on *Bilge Drainage*. Both were listened to with interest, but their strictly technical character precluded the possibility of a lengthy discussion. Mr. Baird's lecture was illustrated by specimens of the fittings described.

The reading of the papers was followed by the report of the Secretary-Treasurer of the Society. He reported as follows:

WASHINGTON, D. C., *Dec. 21, 1891.*

According to Sec. 13 of the By-Laws, I have the honor to make the following report:

The financial condition of the Society is apparent from the itemized list of receipts and expenditures during the year just ended:

RECEIPTS:

Balance on hand January 15, 1891	$1,031.10
Dues and subscriptions for 1891	1,078.51
Dues and subscriptions for 1890	10.00
Dues and subscriptions for 1892	250.00
Sale of JOURNAL	96 55
Miscellaneous	12.01
Advertisements	409.37
Total to date	2,887.54

EXPENDITURES:	
Printing JOURNAL	$1,321.57
Cuts and plates for JOURNAL	304.00
Miscellaneous (postage, stationary, drawings for JOURNAL, &c.)	96.72
Compensation to Secretary-Treasurer	300.00
Total to date	2,022.29
Balance on hand	$865.25
AMOUNTS DUE AND UNPAID:	
Dues for 1891	$50.00
Single copies of JOURNAL, &c.	11.35
Advertisements	154.69
Total	$216.04
AMOUNTS OVERPAID:	
Dues for 1892	$283.75
Advertisements	63.75
Total	$347.50
Actual surplus at end of current year	$733.79

During the past year the number of members, associates and subscribers has increased from about 300 to 363.

A number of amendments to the original By-Laws of the Society, the more important of which were suggested at the last Annual Meeting, were submitted to the members, for vote, and all were adopted.

EMIL THEISS,
Secretary-Treasurer Am. Soc. of Naval Engineers.

After the reading of the report the election of officers was proceeded with. P. A. Engineer G. W. Baird and Asst. Engineer E. Theiss declined to appear as candidates for re-election to the offices held by them during the past year. The vote resulted in the election of—

Chief Engineer David Smith, U. S. Navy, President.

Passed Assistant Engineer W. M. McFarland, Secretary-Treasurer.

Passed Assistant Engineers H. Webster, F. H. Bailey and W. F. Worthington, Members of the Council.

A vote of thanks was tendered to the officers of the past year, to the authors of the papers read, and to the Edison General Electric Company for loan of fittings to illustrate the lecture of P. A. Engineer Baird.

The meeting then adjourned.

BOOK REVIEW.

The D. Van Nostrand Company, of New York, have issued "The Mechanical Engineer's Pocket Book of Tables, Formulæ, Rules and Data," by D. Kinnear Clark.

The name of the well-known engineer whose labors are here brought into convenient shape for reference, is sufficient guarantee of the value of this book.

The arrangement of tables is, in the main, excellent, and the mechanical get-up of the volume is all that can be desired. Its shape is convenient, its type is clear and new, and the index has been carefully compiled. Many of the tables are quite new, and those relating to strengths of materials are of great value.

It is to be regretted, however, that in the experimental data so fully tabulated, English authorities seem to have been consulted rather than recent American practice.

In the electrical portions of this compilation the editor has confined himself almost entirely to English sources of information, and even within these limitations the data compiled is not of the latest available.

For the purposes of American engineers it is unfortunate that the "Imperial" system of wire gauge has been employed throughout. Nevertheless, taken as a whole, this book will undoubtedly be of considerable value to the engineer, both civil and mechanical. Price $3.00.

BY-LAWS.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS.

2. The object of the Society shall be to promote a knowledge of naval engineering by the reading, discussion and publication of papers on professional subjects; by the bringing together of the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in a Journal of the Society; and by a publication of the results of such experimental and other inquiries as may be deemed essential to the advancement of the science.

3. The officers of the Society shall be a President, a Secretary and Treasurer, and a Council, all of whom shall be elected annually.

4. The Society shall be composed of Members, Associates and Honorary Members.

5. Officers of the Engineer and Construction Corps of the Navy, and persons in civil life who were formerly officers in said Corps, shall be eligible as Members.

6. Persons in civil life whose knowledge of engineering is such that they can co-operate with naval engineers in the promotion of professional knowledge, or who are intimately connected with the engineering profession, shall be eligible as Associates.

7. The Secretary of the Navy, the Assistant Secretary of the Navy, the Chief of the Bureau of Steam Engineering of the Navy Department, all ex-Chiefs of the Bureau of Steam Engineering, and such other persons as the Society may elect, shall be Honorary Members.

8. Members shall be admitted upon application upon payment of the annual subscription.

9. An Associate may be admitted upon application in writing, and upon payment of the annual subscription, provided his application have the recommendation of one member, and receive the approval of a majority of the Council.

10. Associates shall be entitled to all the privileges of Members, except voting and holding office.

11. To be eligible as Honorary Member the candidate must receive the unanimous vote of the Council before his name shall be presented to the Society; and the favorable vote of two-thirds of the Members voting shall be necessary for election.

12. The direction and management of the affairs of the Society, and the editing and publishing of the JOURNAL, shall be vested in a Council composed of five Members, two of whom shall be the President and the Secretary and Treasurer. The Council may appoint one of its number Librarian.

13. The President shall preside at the annual meeting, and at the meetings of the Council.

14. The Secretary and Treasurer shall conduct the correspondence of the Society, receive subscriptions, and pay all bills certified by the President. He shall submit an annual statement of the receipts and expenditures of the Society, which shall be audited by three other members of the Council.

15. The Council shall have authority to fill vacancies which may occur during the year.

16. Subscriptions, and all matter intended for publication, shall be sent to the Secretary and Treasurer; but no paper shall be read before the Society at the annual meeting, nor anything published in the JOURNAL, without the approval of the Council.

17. In deciding matters pertaining to the Society, none but Members shall be entitled to a vote.

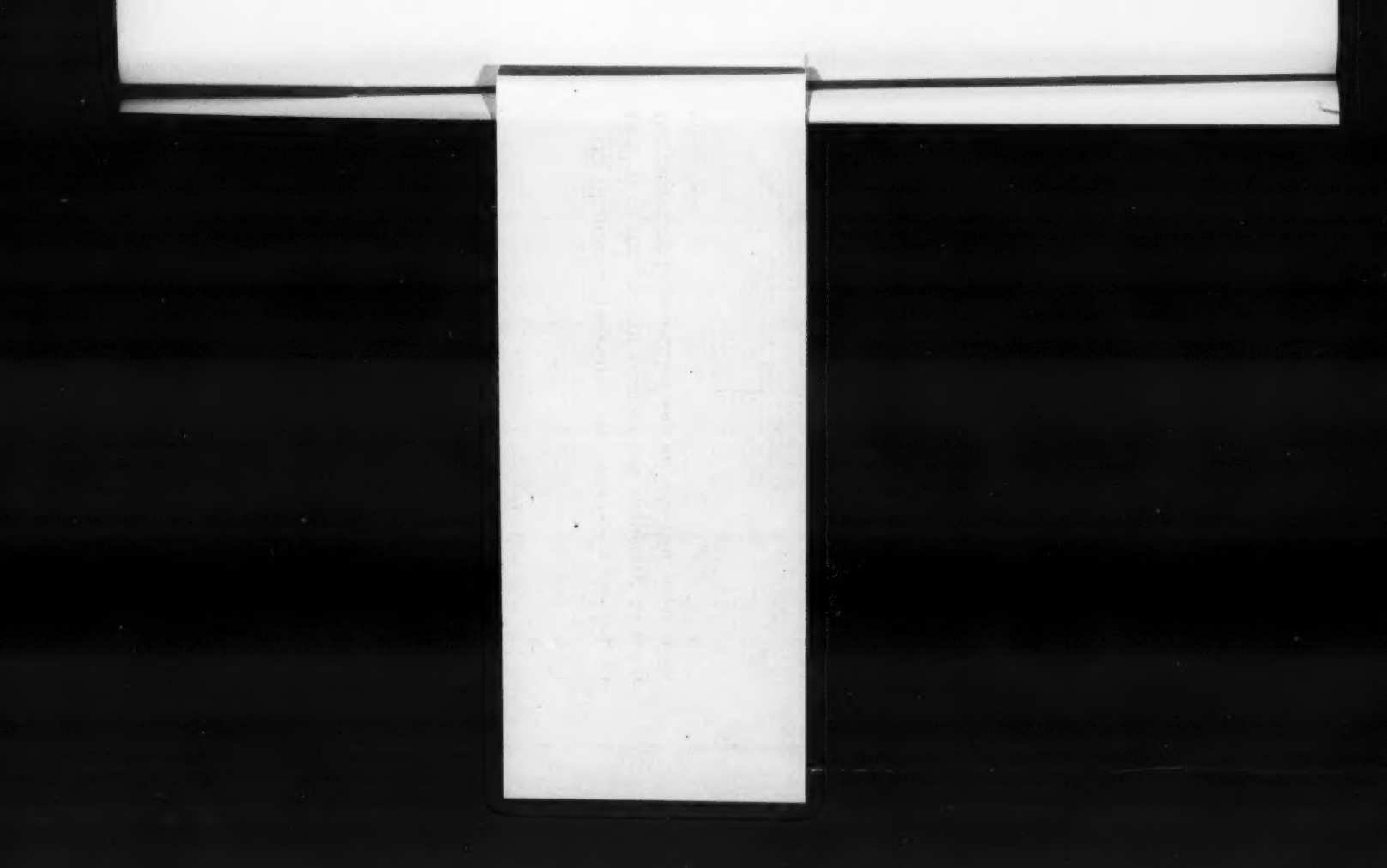
18. The Society shall meet annually in the city of Washington, and at such other places as meetings can be conveniently arranged.

19. The annual subscription shall be five dollars, payable in advance.

20. Copies of the JOURNAL may be sold by the Society at a price which shall be fixed by the Council.

21. Amendments to these rules, or new rules, may be made at any time, provided all the Members are notified of the proposed change, and that a majority of those voting are in favor of the change.

All Members, Associates, and others, in the list appended to the present number are requested to notify the Secretary-Treasurer of any mistakes in name, title, or address.



AMERICAN SOCIETY OF NAVAL ENGINEERS.

OFFICERS FOR 1892.

President:
Chief Engineer **David Smith**, U. S. Navy.

Secretary and Treasurer:
Passed Assistant Engineer **Walter M. McFarland**, U. S. Navy.

Librarian:
Passed Assistant Engineer **H. Webster**, U. S. Navy.

Council:
Chief Engineer **David Smith**, U. S. Navy.
Passed Assistant Engineer **Walter M. McFarland**, U. S. Navy.
Passed Assistant Engineer **H. Webster**, U. S. Navy.
Passed Assistant Engineer **Frank H. Bailey**, U. S. Navy.
Passed Assistant Engineer **Walter F. Worthington**, U. S. Navy.

HONORARY MEMBERS.

(*ex-officio*)

Secretary of the Navy:
The Hon. **B. F. Tracy**, Navy Department.

Assistant Secretary of the Navy:
The Hon. **Jas. R. Soley**, Navy Department.

Chief of Bureau of Steam Engineering:
Geo. W. Melville, Engineer-in-Chief, U. S. N., Navy Department.

Ex-Chiefs of Bureau of Steam Engineering:
Chas. W. Copeland, Consulting Engineer, 24 Park Place, New York.
Chas. H. Haswell, Consulting Engineer, 42 Broadway, New York.
B. F. Isherwood, Chief Engineer, U. S. N. (retired), 111 East 36th street, New York.
Jas. W. King, Chief Engineer, U. S. N. (retired), 3231 Powellton avenue, Philadelphia.
Chas. H. Loring, Chief Engineer, U. S. N. (retired), 239 Clermont avenue, Brooklyn.
Wm. H. Shock, Chief Engineer, U. S. N. (retired), 1404 15th street, Washington, D. C.

MEMBERS.

Able, A. H., Chief Engineer, U. S. N., ... U. S. S. *Newark.*
Addicks, W. R., Superintendent Boston Gaslight Co., ... 24 West street, Boston.
Aldrich, W. S., Associate in Engineering, Johns Hopkins University, ... Baltimore, Md.
Allderdice, W. H., Assistant Engineer, U. S. N., ...
..... Office of Naval Intelligence, Navy Department, Washington, D. C.
Allen, F. B, Vice-President Hartford Steam Boiler Inspection and Insurance Co., Hartford, Conn.
Anderson, M. A., Assistant Engineer, U. S. N., ... Navy Yard, Portsmouth, N. H
Andrade, C., Chief Engineer, U. S. N., ... Wm. Cramp & Sons S. and E. B. Co., Philadelphia, Pa.
Armistead, S. W., Naval Constructor, U. S. N., ... City Point Works, South Boston, Mass
Aston, Ralph, Chief Engineer, U. S. N., ... U. S. S. *Bennington.*
Ayres, S. L. P., Chief Engineer, U. S. N., ... Navy Yard, New York.

Bailey, F. H., Passed Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Baird, G. W., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Dolphin*.
Baker, C. H., Chief Engineer, U. S. N.,.....Navy Yard, Norfolk, Va.
Barnard, G. A.,.....Buckeye Engineering Co., Salem, Ohio.
Barrett, Thos. H , U. S. Local Inspector Steam Vessels, Room 162, P. O. Building, New York.
Barry, J. J., Passed Assistant Engineer, U. S. N., (*retired*), 106 McDonough St., Brooklyn, N. Y.
Barton, J. K., Passed Assistant Engineer, U. S. N., ...U. S. Naval Academy, Annapolis, Md.
Bates, A. B., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Minnesota*, Station G, New York.
Baxter, W. J., Assistant Naval Constructor, ...Navy Yard, Norfolk, Va.
Bayley, W. B , Passed Assistant Engineer, U. S. N.,...U. S. Fish Commission, Washington, D. C.
Beach, E. L., Assistant Engineer, U. S. N.,.....U. S. S. *Philadelphia*.
Bennett, F. M., Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Bevington, M., Assistant Engineer, U. S. N.,.....Homestead Steel Works, Munhall, Pa.
Bieg, F. C., Passed Assistant Engineer, U. S. N., ...U. S. S. *Miantonomoh*.
Blue, Victor, Assistant Engineer, U. S. N.,.....Union Iron Works, San Francisco, Cal.
Borthwick, J. L. D., Chief Engineer, U. S. N.,.....Navy Yard, New York.
Bowers, F. C., Assistant Engineer, U. S. N.,.....U. S. Fish Commission Steamer *Fish Hawk*.
Brecht, T. C.,.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Brooks, W. B., Chief Engineer, U. S. N.,.....
.....Richmond Locomotive and Machine Works, Richmond, Va.
Brown, Jefferson, Chief Engineer, U. S. N.,.....U. S. S. *Vermont*, Navy Yard, New York.
Bryan, B. C., Assistant Engineer, U. S. N.,.....U. S. S. *Bennington*.
Buehler, W. G., Chief Engineer, U. S. N.,.....124 South 17th street, Philadelphia, Pa.
Burd, G. E., Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Burgdorff, T. F., Passed Assistant Engineer, U. S. N., University of Tennessee, Knoxville, Tenn.

Canaga, A. B., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Chicago*.
Carney, Robt. E., Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Carr, C. A., Assistant Engineer, U. S. N.,.....Quintard Iron Works, New York.
Carter, T. F., Assistant Engineer, U. S. N., ... U. S. C. S. S. *Patterson*.
Cathcart, W. L., Consulting Engineer,.....Hoyt, Montgomery County, Pa.
Chambers, W. H., Assistant Engineer, U. S. N.,..... .....U. S. S. *Miantonomoh*.
Cleaver, H. T., Passed Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Collins, Jno. W., Chief Engineer, U. S. R. M., Consulting Engineer,.....
.....Revenue Marine Division, Treasury Department, Washington, D. C.
Conant, F. H., Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Cooley, M. E., Professor Mechanical Engineering,.....University of Michigan, Ann Arbor, Mich.
Cowles, Wm., President Cowles Engineering Company,.....Foot 43d street, South Brooklyn.
Crawford, Robt., Passed Assistant Engineer, U. S. N.,.....328 Front street, Media, Pa.
Creighton, W. H. P., Assistant Engineer, U. S. N.,.....U. S. C. S. S. *Blake*.

Day, W. B., Assistant Engineer, U. S. N.,..... .....Otis Steel Works, Cleveland, Ohio.
Denig, R. G., Passed Assistant Engineer, U. S. N.,.....U. S. Naval Academy, Annapolis, Md.
DeValin, C. E., Chief Engineer, U. S. N.,.....1308 New Hampshire Avenue, Washington, D. C.
Dixon, A. F., Passed Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Doran, Jas. S., Superintending Engineer Inman, International and Red Star S. S. Lines,.....
.....Philadelphia, Pa.
Dowst, F. B., Mechanical Engineer with B. F. Sturtevant Co.,.....Jamaica Plain, Boston, Mass.
Dripps, W. A., Consulting Engineer,.....3324 Walnut street, Philadelphia, Pa.
Dungan, W. W., Chief Engineer, U. S. N.,.....Navy Yard, New York.
Durand, W. F., Professor Marine Engineering and Naval Architecture,.....
.....Cornell University, Ithaca, N. Y.
Dyson, C. W., Assistant Engineer, U. S. N.....Navy Yard, Boston, Mass.

Eckart, W. R., Consulting Engineer,.....217 Sansome street, San Francisco, Cal.
Edson, J. B., Recording Pressure Gauges,.....87 Liberty street, New York.
Edwards, J. R., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Petrel.*
Eldridge, F. H., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Chicago.*
Emanuel, J. M., Passed Assistant Engineer, U. S. N. (retired),.....
.....1810 North 12th street, Philadelphia, Pa.
Emery, C. E., Ph. D., Consulting Engineer,.....915 Bennett Building, New York.
Engard, A. C., Chief Engineer, U. S. N.,.....2131 North 19th street, Philadelphia, Pa.

Farmer, Edward, Chief Engineer, U. S. N.,.....Navy Yard, Portsmouth, N. H.
Ferguson, G. R., M. E. and Pat. Attorney,.....26 South Grove street, East Orange, N. J.
Fisher, Clark, Fisher & Norris, Man'f's of Anvils,.....Trenton, N. J.
Fitch, H. W., Chief Engineer, U. S. N.,.....U. S. Naval Academy, Annapolis, Md.
Fitts, J. H., Professor of Mechanical Engineering,.....Agricultural College, Blacksburg, Va.
Ford, J. D., Chief Engineer, U. S. N.,.....U. S. S. *Alert.*
Freeman, E. R., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Concord.*

Gage, Howard, Passed Assistant Engineer, U. S. N.,.....
.....Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.
Gow, J. L., Passed Assistant Engineer, U. S. N.,.....U. S. Naval Academy, Annapolis, Md.
Greene, A. S., Chief Engineer, U. S. N.,.....Navy Yard, League Island, Pa.
Greene, D. M., Consulting Engineer,.....Troy, N. Y.
Greene, L. R., General Manager Walworth Manufacturing Company,.....
.....35 Concord avenue, Cambridge, Mass.
Green, I. C.,.....Hope Brass Works, P. O. Box 272, Baltimore, Md.
Griffin, R. S., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Philadelphia.*
Grantner, O. C.,.....First Assistant Examiner, Room 124, Patent Office, Washington, D. C.

Habighurst, C. J., Passed Assistant Engineer, U. S. N.,.....
.....Care National Tube Works, McKeesport, Pa.
Hall, Geo. W., Broker,.....115 Broadway, New York.
Hall, H., Assistant Engineer, U. S. N.,.....U. S. S. *Alliance.*
Hall, R. T., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Pensacola.*
Halstead, A. S., Assistant Engineer, U. S. N.,.....U. S. S. *Cushing.*
Handren, John,.....Handren & Robbins, 126 Washington street, New York.
Hannum, J. L., Chief Engineer, U. S. N.,.....U. S. S. *Thetis.*
Harris, W. H., Chief Engineer, U. S. N.,.....U. S. S. *Wabash*, Navy Yard, Boston, Mass.
Hartrath, Armin, Assistant Engineer, U. S. N.,.....Union Iron Works, San Francisco, Cal.
Hasson, W. F. C., Assistant Engineer, U. S. N.,.....U. S. S. *Baltimore*, Union Iron Works.
Hayes, C. H., Assistant Engineer, U. S. N.,.....U. S. S. *Yorktown.*
Heaton, W. W., Chief Engineer, U. S. N.,.....
.....Quintard Iron Works, 12th street and avenue D, New York.
Henderson, A., Chief Engineer, U. S. N.,.....Navy Yard, Boston, Mass.
Herbert, W. C., Assistant Engineer, U. S. N.,.....Wm. Cramp & Sons' S. and E. B. Co., Phila.
Herwig, H., Passed Assistant Engineer, U. S. N.,.....Navy Yard, Washington, D. C.
Hibbs, F. W., Assistant Naval Constructor, U. S. N., University of Glasgow, Glasgow, Scotland.
Higgins, R. B., Assistant Engineer, U. S. N.,.....U. S. S. *San Francisco.*
Hine, R. B., Chief Engineer, U. S. N.,.....U. S. S. *Concord.*
Hogan, T. J.,.....Westinghouse Air Brake Co., P. O. Box 950, Pittsburgh, Pa.
Hollis, I. N., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Charleston.*
Howell, C. P., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Baltimore.*
Hunt, A. M., Assistant Engineer, U. S. N.,.....U. S. F. C. S. *Albatross.*

Inch, Phil., Chief Engineer, U. S. N.,.....U. S. S. *San Francisco.*
Inch, Richard, Passed Assistant Engineer, U. S. N.,.....U. S. S. *Lancaster.*

103

Johnson, G. R., Chief Engineer, U. S. N.,.....1233 New Hampshire avenue, Washington, D. C.
Jones, D. P., Chief Engineer, U. S. N.,.....U. S. S. *Richmond*, Newport, R. I.
Jones, H. W., Assistant Engineer, U. S. N.,.....U. S. S. *Charleston*.

Kaemmerling, G., Assistant Engineer, U. S. N.,.....U. S. C. S. S. *Bache*.
Kafer, J. C., Passed Assistant Engineer, U. S. N., ...
.....Supt'g Engineer Morgan Iron Works, 9th street and E. R., New York.
Kearney, G. H., Passed Assistant Engineer, U. S. N.,.....331 Greene avenue, Brooklyn.
Keilholtz, Pierre O., Sec. and Supt. U. S. Electric Power and Light Co.,.....
.....corner Holliday and Centre streets, Baltimore, Md.
King, C. A. E., Assistant Engineer, U. S. N.,.... Col. Iron Works and D. D. Co., Baltimore, Md.
King, W. R., Passed Assistant Engineer, U. S. N. (retired),.....Asheville, N. C.
Kinkaid, T. W., Assistant Engineer, U. S. N.,.....Wm. Cramp and Sons' S. and E. B. Co., Phila.
Kirby, A., Chief Engineer, U. S. N.,.....Navy Yard, Washington, D. C.
Koester, O. W., Assistant Engineer, U. S. N.,.....U. S. S. *Essex*.
Kutz, G. F., Chief Engineer, U. S. N.,.....Union Iron Works, San Francisco, Cal.

Lawa, E., Chief Engineer, U. S. N., ...U. S. S. *Pensacola*.
Leavitt, E. D., Consulting Engineer,.....2 Central Square, Cambridgeport, Mass.
Leitch, R. R., Passed Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Leonard, J. C., Assistant Engineer, U. S. N.,.....U. S. C. S. S. *McArthur*.
Leonard, S. H., Jr., Assistant Engineer, U. S. N.,.....Bath Iron Works, Bath, Me.
Leopold, H. G., Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Little, W. N., Passed Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Lloyd, John,.....558 Water street, New York.
Lowe, J., Chief Engineer, U. S. N.,..... .Wm. Cramp & Sons' S. and E. B. Co., Philadelphia.

Maccarty, G. M. L., Chief Engineer, U. S. N.,.....Bath Iron Works, Bath, Me.
Macomb, D. B., Chief Engineer, U. S. N.,.....28 Arlington street, Cambridge, Mass.
Magee, E. A., Chief Engineer, U. S. N.,.....
.....Board of Inspection of Merchant Steamships, Postoffice Building, New York.
Magee, G. W., Chief Engineer, U. S. N.,.....U. S. S. *Miantonomoh*.
Main, H., Passed Assistant Engineer, U. S. N., ...Care Navy Department, Washington, D. C.
Manning, C. H., Passed Assistant Engineer, U. S. N. (retired),.....
.....Supt. Amoskeag Mills, Manchester, N. H.
Matthews, C. H., Assistant Engineer, U. S. N.,.....Bolton House, Harrisburg, Pa.
Mattice, A. M., Consulting Engineer,.....2 Central Square, Cambridgeport, Mass.

McAllister, A., Assistant Engineer, U. S. N.,..... .....Navy Yard, New York.
McAlpine, K., Assistant Engineer, U. S. N.,.....Navy Yard, Norfolk, Va.
McCartney, D. P., Chief Engineer, U. S. N.,..... S. L. Moore & Sons, Elizabethport, N. J.
McCutcheon, J. F., Chief Engineer City Hall,.....City Hall, Philadelphia, Pa.
McElmell, J., Chief Engineer, U. S. N.,.....
.....President Board of Examiners, Postoffice Building, Philadelphia, Pa.
McElnell, Thomas, Agent Vacuum Oil Co.,..... .....1931 Spring Garden street, Philadelphia, Pa.
McElroy, G. W., Passed Assistant Engineer, U. S. N.,.....U. S. S. *San Francisco*.
McEwan, H. D., Chief Engineer, U. S. N.,.....
.....Member Board of Examiners, Postoffice Building, Philadelphia, Pa.
McFarland, W. M., Passed Assistant Engineer, U. S N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
McKean, F. G., Chief Engineer, U. S. N.,.....U. S. S. *Boston*.
McKean, J. S., Assistant Engineer, U. S. N.,.....U. S. S. *Chicago*.
McNary, I. R., Chief Engineer, U. S. N.,.....Navy Yard, Portsmouth, N. H.
Mickley, J. P., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Fern*.
Milligan, R. W., Passed Assistant Engineer, U. S. N.,.....Navy Yard, Mare Island, Cal.

147

Miner, L. D., Assistant Engineer, U. S. N.,.....U. S. S. *Newark*.
Moore, J. W., Chief Engineer, U. S. N.,.....Navy Yard, Mare Island, Cal.
Moore, W. S., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Vesuvius*.
Morgan, Leo.,.....Union Iron Works, San Francisco, Cal.
Moritz, Albert, Assistant Engineer, U. S. N.,.....
.....Quintard Iron Works, 12th street and Avenue D, New York.
Morley, A. W., Chief Engineer, U. S. N.,.....Quintard Iron Works, New York.

Nagle, C. F., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Michigan*.
Naumann, W. H., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Philadelphia*.
Nones, H. B., Chief Engineer, U. S. N.,.....
.....Member Board of Examiners, Postoffice Building, Philadelphia, Pa.
Norton, H. P, Assistant Engineer, U. S. N.,.....U. S. S. *Concord*.

Offley, C. N., Assistant Engineer, U. S. N.,.....Navy Yard, New York.
Ogden, J. S., Passed Assistant Engineer, U. S. N.,.....Navy Department, Washington, D. C.

Parks, W. M., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Atlanta*.
Pemberton, J., Passed Assistant Engineer, U. S. N. (retired), Care State College, Center Co., Pa.
Perry, J. H., Passed Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Pickrell, J. M., Assistant Engineer, U. S. N.,.....
.....Richmond Locomotive and Machine Works, Richmond, Va.
Potts, Stacey, Passed Assistant Engineer, U. S. N.,.....U. S. S. *Newark*.

Rae, C. W., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Atlanta*.
Ransom, G. B., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Pinta*.
Rearick, P. A., Chief Engineer, U. S. N.,.....U. S. S. *Iroquois*.
Redgrave, DeW. C., Assistant Engineer, U. S. N.,.....
.....Col. Iron Works and D. D. Co., Baltimore, Md.
Reid, R. I., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Newark*.
Rhoades, H. E., Assistant Engineer, U. S. N., (retired), ...Mount Vernon, New York.
Roberts, E. E., Prop. Roberts' Boiler,.....18 Cortlandt street, New York.
Robie, E. D., Chief Engineer, U. S. N., ...
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Robinson, L. W., Chief Engineer, U. S. N.,.....
.....Chief of Bureau of Machinery, World's Columbian Exposition, Chicago, Ill.
Roche, G. W., Chief Engineer, U. S. N.,.....Col. Iron Works and D. D. Co., Baltimore, Md.
Roelker, C. R., Chief Engineer, U. S. N.,.....Navy Yard, Norfolk, Va.
Rommell, C. E., Assistant Engineer, U. S. N.,.....U. S. S. *Yorktown*.
Ross, H. S., Chief Engineer, U. S. N.,.....U. S. S. *Essex*.
Rowbotham, W., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Baltimore*.

Salisbury, G. R., Assistant Engineer, U. S. N.,.....Care Navy Department, Washington, D. C.
Sampson, B. C., Assistant Engineer, U. S. N.,.....U. S. Naval Academy, Annapolis, Md.
Schell, F. J., Passed Assistant Engineer, U. S. N.,.....Naval Academy, Annapolis, Md.
Scribner, E. H., Assistant Engineer, U. S. N.,.....U. S. S. *Boston*.
Selden, W. C., Superintending Engineer Clyde Line Steamers,.....Pier 29, East River, New York.
Sims, Gardiner C., General Manager Armington & Sims Engine Co.,.....Providence, R. I.
Sloane, J. D., Assistant Engineer, U. S. N. (retired),.....
.....U. S. Supervising Inspector Steam Vessels, St. Paul, Minn.
Smith, David, Chief Engineer, U. S. N., Member Steel Inspection Board,.....
.....Navy Department, Washington, D. C.
Smith, J. A. B., Chief Engineer, U. S. N.,.....Col. Iron Works and D. D. Co., Baltimore, Md.
Smith, S. L., Supt. Roxbury Carpet Co.,.....37 Simmons street, Boston, Mass.
Smith, W. Strother, Assistant Engineer, U. S. N.,.....Navy Yard, Norfolk, Va.

Smith, W. S , Chief Engineer, U. S. N.,.....U. S. S. *Marion*.
Smith, W. Stuart, Assistant Engineer, U. S. N. (retired),.....Yokohama, Japan.
Spangler, H. W., Professor Dynamic Engineering,.....University of Pennsylvania, Philadelphia.
Stevenson, H. N., Passed Assistant Engineer, U. S. N., ...U. S. S. *Alliance*.
Stickney, H. O., Assistant Engineer, U. S. N.,.....U. S. S. *Bennington*.
Strickland, G. D., Passed Assistant Engineer, U. S. N.,....Standard Steel Works, Thurlow, Pa.

Taylor, D. W., Naval Constructor, U. S. N.,Wm. Cramp & Sons' S. and E. B. Co.,Philadelphia.
Taylor, R. D., Passed Assistant Engineer, U. S. N.,.....U. S. Ironclads, Richmond, Va.
Tawresey, J. G., Assistant Naval Constructor, U. S. N.,.....Navy Yard, New York.
Theiss, Emil, Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Thomson, J. W., Chief Engineer, U. S. N., Wm. Cramp & Sons' S. and E. B. Co., Philadelphia.
Tobin, J. A., Passed Assistant Engineer, U. S. N. (retired),.....19 Cliff street, New York.
Tower, G. E., Chief Engineer, U. S. N.....U. S. S. *Kearsarge*.
Towne, N. P., Chief Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Trilley, Joseph, Chief Engineer, U. S. N.,.....Union Iron Works, San Francisco, Cal.

Van Buren, J. D., Consulting Engineer,.....Newburgh, N. Y.

Varney, W. H., Naval Constructor, U. S. N.,....Col. Iron Works and D. D. Co., Baltimore, Md.
Voorhees, P. R., Counsellor-at-Law,.....32 Nassau street, New York.

Warburton, E. T., Passed Assistant Engineer, U. S. N.,.....U. S. S. *San Francisco*.
Warren, B. H.,.....Yale & Towne Manufacturing Co., Stamford, Conn.
Weaver, W. D., Electrical Engineer,.....Hall Building, Troy, N. Y.
Webster, H., Passed Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Wharton, B. B. H., Chief Engineer, U. S. N.,.....U. S. S. *Lancaster*.
White, W. W., Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Whittaker, E. J., Chief Engineer, U. S. N.,.....U. S. S. *Philadelphia*.
Williamson, J. D.,. ....
.....Williamson Bros.' E. and B. Works, York and Richmond streets, Philadelphia, Pa.
Williamson, Thom, Chief Engineer, U. S. N.,.....
.....Supt. State, War and Navy Department Building, Navy Department, Washington, D. C.
Willis, C. C., Assistant Engineer, U. S. N. (retired),.....New Market, Tenn.
Willits, A. B., Passed Assistant Engineer, U. S. N.,.....
.....Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.
Willits, Geo. S., Passed Assistant Engineer, U. S. N.,.....U. S. S. *Boston*.
Wilmer, J. R. Passed Assistant Engineer, U. S. N., Care Navy Department, Washington, D. C.
Wilson, F. A., Chief Engineer, U. S. N.,.....U. S. S. *Charleston*.
Winchell, W. P., Assistant Engineer, U. S. N.,.....U. S. S. *Kearsarge*.
Windsor, W. A., Chief Engineer, U. S. N.,.....Iowa Iron Works, Dubuque, Iowa.
Wood, B. F., Chief Engineer, U. S. N.,.....Continental Iron Works, Station G, Brooklyn.
Wood, J. L., Assistant Engineer, U. S. N.,.....U. S. S. *Marion*.
Woods, A. T., Professor Mechanical Engineering,.....Washington University, St. Louis, Mo.
Woodward, J. J., Naval Constructor, U. S. N.,.....
.....Bureau Construction and Repair, Navy Department, Washington, D. C.
Worthington, W. F., Passed Assistant Engineer, U. S. N.,.....
.....Bureau Steam Engineering, Navy Department, Washington, D. C.

Zeller, Theo., Chief Engineer, U. S. N.,.....15 West 12th street, New York.
Zinnell, G. F., Professor of Engineering,.....
.....Spring Garden Institute, 816 North 26th street, Philadelphia, Pa.

ASSOCIATES.

Angstrom, Arend, Suptd'g Engineer Cleveland Shipbuilding Co.,.....Cleveland, Ohio.

Babcock, W. I., Manager Chicago Shipbuilding Co.,.....
.....101st street and Calumet River, Colehouse, Ill.
Bailey, W. H., Agent American Tube Works,.....20 Gold street, New York.
Beavor-Webb, J., Naval Architect,.....45 Broadway, New York
Belcher, A. W., Superintendent Machine Shops Cornell Steamboat Co.,.....Rondout, N. Y.
Bissell, G. W., Mechanical Engineer,.....Box 327, Ames, Ia.
Blauvelt, Albert, Marine Electrician,.....
.....Edison General Electric Co., 18 West 9th street, New York.
Blomberg, C. A.,.....Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.
Bolten, A......850 Market street, San Francisco, Cal.
Bonneville, A. A......77 Liberty street, New York.
Boyd, J. T., Suptd'g Engineer Atlantic Works,.....East Boston, Mass.
Bray, Chas. D, Prof. Engineering, Tuft's College,.....Somerville, Mass.

Calder, C. B., Suptd'g Engineer Menominee Transit Co.,.....Cleveland, Ohio.
Coryell, Miers, Consulting Engineer,.....21 East 21st street, New York.
Cox, Irving, Marine Architect,.....50 Wall street, New York.
Cramp, B. H., B. H. Cramp & Co., Brass Founders, York and Thomson streets, Philadelphia, Pa.
Cramp, C. D., with B. H. Cramp & Co, Brass Founders,.....
.....York and Thompson streets, Philadelphia, Pa.
Cramp, E. S., Suptd'g Engineer Wm. Cramp & Sons' S. and E. B. Co.,.....Philadelphia, Pa.
Cust, Leopold,.....3 Park Terrace, Govan, Glasgow, Scotland.

Davidson, M. T., Proprietor Davidson Steam Pump Works,.....49 Keap street, Brooklyn, N. Y.
Davis, L. D., Builder of Steam Yachts and Launches,.....Davis-Farrar Co., Erie, Pa.
DeRycke, Jos., Mechanical Engineer,.....145 Broadway, New York.
Dickinson, R. T., Naval Architect,.....Delaware River Iron Works, Chester, Pa.
Dow, Geo. E., Prop. Dow Steam Pump Works,.....114 Beale street, San Francisco, Cal.
DuBosque, F. L., Assistant Superintendent Motive Power Penna. R. R.....Jersey City, N. J.

Elliott, Wm. E,.....Goodrich Transp. Co., Manitowoc, Wis.

Fehrenbatch, John, Supervising Inspector Steam Vessels, Seventh District,.....Cincinnati, O.
Field, C. J.,.....Field Engineering Co., 15 Cortlandt street, New York.
Flach, Emil, Chief Draughtsman Sam'l L Moore & Sons' Co.,.....Elizabethport, N. J.
Fletcher, Andrew, Jr.,..... W. & A. Fletcher, 157 West 73d street, New York.
Forsyth, Robert, Assistant Manager Union Iron Works,.....San Francisco, Cal.
Frear, H. P., Naval Architect,.....Union Iron Works, San Francisco, Cal.
French, D. M., First Assistant Engineer, U. S. R. M.,.....68 Royal street, New Orleans, La.
Frerichs, J. A, Marine Engineer,.....Osterholz-Schwarmbeck by Bremen, Germany.

Griffith, Edwin, Chief Designer Fairfield Ship and Engine Building Works,.....
.....2 Glenavon Terrace, Partick, Glasgow, Scotland.
Gunderson, A., Chief Draughtsman Handren & Robbins,.....126 Washington street, New York.

Hall, Geo. D., General Manager Wainwright Manufacturing Co.,.....Boston, Mass.
Henderson, A. P.,.....Box 398, Newport News, Va.
Henning, Gus, Consulting Engineer,.....762 Temple Court, 5 Beekman street, New York.
Henthorn, J. T.,.....Remington & Henthorn, 146 Westminster street, Providence, R. I.
Hoffmire, John D., Ship and Steamboat Joiner,.....808 5th street, New York.
Howell, W. N.,.....Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.
Hyde, Chas. E., Suptd'g Engineer Bath Iron Works,.....Bath, Me.
Hyde, John S.,.....Bath Iron Works, Bath, Me.

Janson, E. N.,.....Globe Iron Works, Cleveland, Ohio.
Jefferis, J. E., Chief Engineer, U. S. R. M.,.....U. S. R. S. *Corwin*, San Francisco, Cal.
Jordan, S. S.,..... Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.

Katzenstein, L.,.....L. Katzenstein & Co., 357 West street, New York.
Kerlin, J. M. Sharpless, Mechanical Engineer,..... Elwyn, Pa.
King, F. B., Superintendent Marine Department Pennsylvania Steel Co.'s Works,.....
.....Sparrows Point, Md.
Kirby, F. E., Suptd'g Engineer Detroit Drydock Co.,.....Detroit, Michigan.
Kunstadter, J. J., Marine Engineer,.....31 Broadway, New York.

Laval, Geo. de, Chief Draughtsman Knowles Steam Pump Works,.....Warren, Mass.
Lincoln, J. M., Assistant Suptd'g Engineer Clyde Steamship Co.,.....
.....Pier 29, East River, New York.
Loring, H. Jr., General Manager City Point Works,.....South Boston, Mass.
Lundborg, C. G., Naval Architect,.....430 West 14th street, New York.

Mahon, Wm. L. E., Consulting and Mechanical Engineer, American Steel Barge Co.,.....
.....West Superior, Wis.
Maschmeyer, A. M. P.,.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Matthews, E. M.,.....Wm. Cramp & Sons' S. and E. B. Co., Philadelphia, Pa.
McAllister, C. A.,.....Bureau Steam Engineering, Navy Department, Washington, D. C.
McMakin, Jos., Suptd'g Engineer Columbian Iron Works and D. D. Co., ...Baltimore, Md.
Mellen, C. J., Chief Draughtsman Richmond Locomotive and Machine Works,...Richmond, Va.
Miller, Walter, Chief Draughtsman and Engineer Globe Iron Works,.....Cleveland, Ohio.
Mills, A. B., Coppersmith,.....127 Avenue D, New York.
Mooney, Thos., Chief Draughtsman Morgan Iron Works, 9th street and East River, New York.
Mosher, C. D., Naval Architect and Marine Engineer,..... 19 Whitehall street, New York.

Nystedt, Thure, Marine Ingeniörs Statens Chefs Expedition,.....Stockholm, Sweden.

Palmer, N. F., President Quintard Iron Works,.....12th street and Avenue D, New York.
Pankhurst, J. F., Vice-President and General Manager Globe Iron Works,.....Cleveland, Ohio.
Parsons, H. deB., Consulting Engineer,.....32 William street, New York.
Pray, Thos. Jr., Consulting Engineer,.....Room 193 Times Building, New York.
Pusey, C. W., General Manager Pusey & Jones Co.,.....Wilmington, Del.

Quintard, G W., Partner Quintard Iron Works,.....12th street and Avenue D, New York.

Raynal, A. H., Superintending Engineer Sam'l L. Moore & Sons Co.,.....Elizabethport, N. J.
Robinson, E. P., Manufacturer Boilers,..... 404 Border street, East Boston, Mass.
Rowland, Jonathan, Vice President Morris, Tasker & Co., 224 South 3d street, Philadelphia, Pa.
Rowland, T. F. Jr., Secretary and Treasurer Continental Iron Works, Station G, Brooklyn, N. Y.
Rozycki, Stephen,.....Bureau Steam Engineering, Navy Department, Washington, D. C.

Sanderson, R. P. C., Superintendent of Motive Power N. & W. Ry., ...Roanoke, Va.
Sewell, W. T., Chief Engineer R. M. S. S *Majestic*,.....
.....Care White Star Line, office 41 Broadway, New York.
Schwartz, E. G., First Assistant Engineer U. S. R. M ,.....
.....U. S. R. S. *Manhattan*, Barge Office, New York.
Scott, Irving M., Vice President and General Manager Union Iron Works, San Francisco, Cal.
See, Horace, Marine Engineer and Naval Architect,.....1 Broadway, New York.
Shaw, T. Jackson, Suptd'g Engineer Harlan & Hollingsworth Co.,.....Wilmington, Del.
Slayton, H. O., 2d Assistant Engineer, U. S. R. M.,.....U. S. R. S. *Galveston*
Spiers, James, Mechanical Engineer, Fulton Iron Works, 213 Fremont street, San Francisco, Cal.

96

Steers, Henry, Naval Architect,.....111th Ward Bank, New York.
Sylven, Walfrid, Suptd'g Engineer Marine Construction Light House Board,.....
.....Treasury Department, Washington, D. C.

Tabor, Harris, Superintendent Tabor Manufacturing Co.,.....21 Elm street, Elizabeth, N. J.
Taylor, Stevenson, Vice-President and General Superintendent of W. & A. Fletcher Co.,.....
.....12th and Hudson streets, Hoboken, N. J.
Tucker, Edwin W., Consulting and Suptd'g Engineer,.....
.....Care Risdon Iron Works, San Francisco, Cal.

Varney, W. W.,.....Col. Iron Works and D. D. Co., Balto., Md.
Volz, W. E, Suptd'g Engineer South Brooklyn Steam Engine Works, ...Brooklyn, N. Y.
Voss, Ernst, Mechanical Engineer, Blohm & Voss,.....Hamburgh, Germany.

Ward, Charles, Proprietor Ward Boiler, ...Charleston, Kanawha Co., W. Va.
Warrington, J. N., Secretary Vulcan Iron Works,.....Chicago, Ill.
Waschmann, Wm.,.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Weese, George,..... 97 Byrne's Roads, Partick, Glasgow, Scotland.
Wheeler, F. M, Mechanical Engineer and General Manager Geo. F. Blake Manf. Co ,.....
.....93 Liberty street, New York.
Wilcox, S., Vice President Babcock & Wilcox Co.,.....30 Cortlandt street, New York.
Williams, S. M.,..... .....13 Rutlandt Place, Govan, Glasgow, Scotland.

Yarrow, A. F., Naval Architect and Mechanical Engineer, ... Isle of Dogs, Poplar, London.

EXCHANGES.

American Shipbuilder,.....115 Broad street, New York.
American Society of Mechanical Engineers,.....12 West 31st street, New York.
Army and Navy Journal,.....240 Broadway, New York.

Cassier's Magazine,.....Potter Building, New York.

Franklin Institute,.....Philadelphia, Pa.

Institution of Civil Engineers,.....25 Great George street, Westminster, London, S. W.

Marine Journal,.....132 Nassau street, New York.
"Mechanics,".....430 Walnut street, Philadelphia, Pa.

North East Coast Institute of Engineers and Shipbuilders,.....
.....Newcastle-upon-Tyne, England.
Naval Institute,..... Naval Academy, Annapolis, Md.

Railroad and Engineering Journal,.....Postoffice Box 1515, New York.
Rowell, Geo. P. & Co., "Printer's Ink,".....10 Spruce street, New York.

Stevens Indicator,.....Stevens Institute, Hoboken, N. J.

United Service,.....1510 Chestnut street, Philadelphia, Pa.

Zeitschrift des Vereines Deutscher Ingenieure,.....131 Potsdamer street, Berlin.

SUBSCRIBERS.

Ayer, N. W. & Son, Advertising Agents,.....800 Chestnut street, Philadelphia, Pa.

Barber, F. M , Commander, U. S. N.,.....U. S. S. *Monocacy.*
Blake Manufacturing Co.,.....East Cambridge, Mass.
Boston Public Library,..... ....Boston, Mass.
Bureau Steam Engineering Library,.....Navy Department, Washington, D. C.

Gowen, F. H.,..... .....513 Case avenue, Cleveland, Ohio.

Hanford, F., Lieutenant Commander, U. S. N.,..... .....Scottsville, N Y.
Hanscom, C.,..... .....144 Hawthorn street, Chelsea, Mass.
Hibbard, Thos., Treasurer George Lawley & Son Corporation,.....South Boston, Mass.

Jones, Geo. W., Bookseller,.....Annapolis, Md.

Kendall, Edw. & Sons,.....Charles River Iron Works, 72 Main street, Cambridgeport, Mass.
Kuehl, W. H., Bookseller,.....73 Jaeger street, Berlin, W.

Langley, Gerald, Captain, R. N., Naval Attaché British Legation, Washington,.....
.....1015 Connecticut avenue, Washington, D. C.

Manning, Maxwell & Moore,..... ....111 Liberty street, New York.

Navy Department Library,.....Navy Department, Washington, D. C.

Nakamura, Shizuka; Lieutenant I. J. N., Naval Attaché Japanese Legation, Washington, D. C.

Patent Office Library, England,.....25 Southampton Buildings, London, W. C.
Prindle, H. B. & Co., Manufacturers' Advertising Bureau,.....
.....Exchange Building, 53 State street, Boston, Mass.

Record of American and Foreign Shipping,.....37 William street, New York.

Stanton, J. R., Paymaster, U. S. N.,.....for Naval Lyceum, Navy Yard, Mare Island, Cal.

Tanquet, A., Librarian Ecôle du Génie Maritime,.....Paris, France.

United Service Club,.....Washington, D. C.

Western, B. R., Manufacturers' Advertising Bureau and Press Agency, ...
.....111 Liberty street, New York.
Williams, J. A,..... .....Box 40, Newport News, Va.